Heating · Piping and Air Conditioning

Vol. 10, No. 2 • *February, 1938*

Control Systems for Oil Fired Boilers

Recent Practice in Industrial Installations Described

• *By* Paul R. Unger* and Kalman Steiner* •

Since a general discussion of combustion control systems for oil fired boilers last appeared in these pages there have been a number of refinements and improvements in both equipment and methods of application. . . . . The authors discuss in some detail several installations which illustrate latest practices

SINCE the authors last discussed the general subject of combustion control systems in these pages,[1] a number of refinements and improvements have been developed in both equipment and methods of application. The scope and utility of these can perhaps best be illustrated by reviewing in some detail two or three installations in a manner which will reveal how actual plants were planned and erected during the past year. This discussion will bring out the two principal aims—to secure the maximum combustion efficiency and the greatest possible safeguard against unsafe operation or interruption of service.

Control for a 150 Hp Boiler

One such plant is the recently completed Chicago factory of the National Adhesives Co. Briefly, the boiler room equipment comprises a 150 hp H.R.T. boiler operating at about 125 lb pressure, oil fired. It is known that load variations will range from 30 to 250 hp, or from 20 to 167 per cent rating. The fuel used is variously known as liquid asphalt, road tar, bunker C, or heavy No. 6 fuel oil. The boiler plant was to be entirely automatic, both day and night, but was to operate with as continuous a fire as possible—that is, the number of starts and stops was to be held to a minimum.

The original thought was to use an electrical modulating control system, to be set at 125 lb and to graduate the fire so that it would become smaller or larger in response to load conditions or steam pressure deviation from that point, in conjunction with a set of intermittent controls to stop the burner completely at 130 lb and restart it at 120 lb. This arrangement was discarded, however, after preliminary study indicated that its use would result in frequent intermittent operations and that furthermore the modulating system would cause the burner to assume full size immediately, since the starting pressure of 120 lb is below that which the modulating system would be required to maintain. Instead, it was decided to stop burner operation at 135 lb (should the smallest fire possible under the modulating control drive steam pressure to that point), and to restart the burner at a boiler pressure of 130 lb. This causes the burner to restart on a small fire and to increase gradually as load conditions require, thereby allowing the burner more opportunity to find a position or firing rate commensurate with the load.

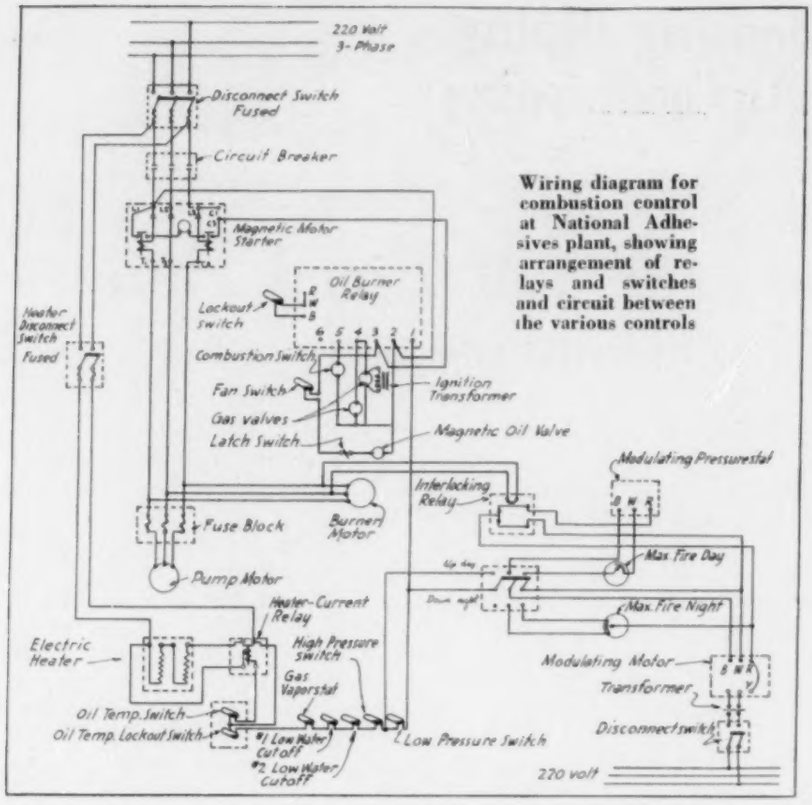
The next step in developing this control system was to provide for complete closing of the air ports each time the burner stops, in order to minimize loss of heat by passage of air through the firebox during the off period. This is accomplished by the use of a relay which is closed whenever the main motor starting relay is opened—the circuit being hooked up so that these contacts drive the modulating motor, and therefore the air dampers which it controls, to a completely closed position. Hence the burner will always start with a small fire, regardless of what the boiler pressure may be at the moment of starting. The fastest cycle of the modulation motor selected is two minutes, so that even with abnormally low steam pressure a two minute period must elapse before full firing rate is reached. The purpose, of course, is thereby to assure smooth and puffless starting, under all conditions. Normally, of course, the boiler pressure at the moment of starting could not be less than 130 lb except at the initial start up in the morning, or after the burner has been stopped manually.

Twin Ignition Provided

To further safeguard smooth starting, twin ignition is provided; that is, there are two pilots, one on each side of the burner nozzle. Both pilots are adequate to light the burner singly, should one or the other fail. One pilot is of the expanding gas type, and carries a small flame continuously, and expands it to full size each time the burner starts. The other is electric-gas, being capable of igniting itself from an electric spark whenever needed. Even on this one, however, a small pilot is kept burning. The spark therefore serves merely to light the gas should the gas flame be completely extinguished. The gas supply pipe serving these two pilots carries a lockout switch in the form of a manually reset vaporstat. Any failure in gas pressure will cause the vaporstat to open the burner control circuit, and to keep it open until manually reset. The thought here is that the cause of the gas failure

*Mid-West Heat Service Co.

[1] HEATING, PIPING & AIR CONDITIONING, October, 1936.



Wiring diagram for combustion control at National Adhesives plant, showing arrangement of relays and switches and circuit between the various controls

should be investigated before normal operation is resumed.

Still one more device is provided in the interest of smooth starting, in the form of another lockout switch attached to the modulating motor lever. This switch permits the burner to start only when the modulating motor is in minimum position. Should the modulating motor move out of this position for any reason whatever, while the burner is down, the burner cannot restart until the modulating motor is brought back to minimum. With equipment as large as this burner, capable of burning over 100 gallons of oil per hour, a bad puffback might result should the burner attempt to start with a larger quantity of oil flowing.

So far, it might seem that our principal concern has been that of safety, and of course this is a very important aspect of boiler operation; in fact, there are still more safety devices on this boiler, yet to be discussed. But it must be brought out here that the modulating system is closely related to efficiency as well as to safety.

It has already been mentioned that the modulating equipment has been utilized to close off air flow through the boiler during shut downs. But while the burner is in operation the modulating equipment gives very close control also over combustion efficiency, for the reason that for every change in the oil firing rate, however minute, there is a corresponding change made in the air flow rates, both primary and secondary. Thus it is possible to hold excess air at a minimum, with consequent high CO_2 content and equivalent high combustion efficiency. The modulating equipment, by the way, is assisted in controlling secondary air by means of a balanced draft regulator installed on the smoke uptake.

Discussion of the remaining safety controls can now be completed. A low water cutout switch on the boiler water level controller stops the burner whenever the water level falls below the safe level. A separate low water cutout is set about 2 in. lower, and is wired in series, for additional safety. In addition, the water level controller carries a switch to ring an alarm bell in the factory proper, as a warning that the water level is unsafe. A combustion safety switch shuts down the burner should ignition fail at time of starting, or should oil flow or flame cease at any time the burner is running. A latch switch closes the magnetic oil valve whenever the burner is swung away from the correct firing position. A fan switch closes the magnetic oil valve should the fan fail to develop adequate air pressure to carry oil away from the nozzle. The electric oil heater thermostat also carries a circuit which will lock out the burner should the oil temperature fall below a point deemed minimum for safe operation. Finally, a manually operated double throw switch will change the burner control over to low pressure for nights and week ends, etc., but this switch is locked into position when the engineer leaves the premises, preventing operation of the boiler at high pressure; the engineer alone can unlock it and throw it to high pressure control. During the night, therefore, or at any other time when the load consists only of heating the building, the boiler operates merely as a low pressure heating boiler.

Entirely aside from the strictly oil burner controls so far enumerated, the installation includes the customary electrical devices of overload relays on the motor starters, fuses against short circuits, and also a circuit breaker which acts on both short circuit and motor overload.

Controlling Oil Heaters

It has already been mentioned that the electric oil heater is equipped with a thermostat. This is set to cut off the electric current at an oil temperature of 130 F. The steam operated oil heater, which heats the fuel oil at all times except starting up periods, is likewise equipped with a temperature regulator consisting of a thermostatic steam valve operated from a bulb immersed in the oil leaving the heater. The temperature setting on this control throttles the steam as the oil temperature rises, and is intended to stop completely steam flow into the heater at an oil temperature of 160. This arrangement of temperature regulation prevents the electric heater from overlapping into the range of the steam heater. Only when steam is not available, and the oil temperature consequently falls to below 130, will the electric heater current be switched on.

Another interesting example of modern control is to

be found in the newly erected factory building of the Lady Esther Co. in the Clearing industrial district of Chicago. Here the oil burners are utilized solely for building heating. The heating system itself is of the forced hot water type, employing mostly direct radiation but also using about 50 unit heaters. There are two boilers—125 hp Scotch marines—each fired by a horizontal rotary type oil burner. Here, too, the fuel is bunker C.

Burner Control for Hot Water Heating System

The preheating is principally accomplished by steam secured from a small high pressure boiler which supplies steam for processes in the factory. But a heating coil in the tank is supplied with hot water from the heating boilers by means of a small circulating pump, which pump is controlled from an immersion thermostat inserted in the oil suction from the tank. Finally, an electric heater is also on hand to warm up the oil should there be neither steam nor hot water.

The safety controls for these burners include high limit aquastats in each boiler, corresponding to high pressure limit switches on steam boilers, as well as the customary stack switches, latch switch and fan switch on each burner. The main operating controller is a potentiometer type aquastat immersed in the main hot water stream leaving the boiler header. This aquastat is of the double circuit type and directly controls the two modulating motors—one on each burner. As the water temperature begins to exceed the point at which the aquastat is set to start modulating, the flame size is reduced, and with it the draft is lowered by turning of the uptake damper through a linkage of levers. The water temperature at which modulation begins is instantly adjustable on the potentiometer aquastat by the turning of a screw.

This arrangement enables the engineer to select whatever water temperature he cares to maintain in his heating system, his choice being predicated of course on the outside temperature. It is also possible to connect the aquastat to an outdoor controller, which in its turn could automatically cause the aquastat to select a water temperature suited to any prevailing outdoor condition.

It is interesting to note that the complete wiring diagram for this installation calls for four different electric voltages. All the motors are operated on 440 volt, 3 phase power, which is the power supplied throughout the factory. The electric oil heater, however, despite its load of 5 kw or about 7.5 hp, is carried on a special 220 volt, single phase line because of the hazard of insulation rupture in the heating elements immersed in fuel oil should 440 volts be impressed across them. The small motor of the hot water circulator to the tank coil is run on 110 volts taken from the lighting system. The modulating motors are run at 20 volts, the current source being transformers hooked onto the 110 volts lighting.

Because the motor starting switches for the various motors are centrally located on a common panel, and because of the distance between the boiler fronts and the panel location, the motor starters for the burner motors are wired to remote push button stations placed at the respective boiler fronts, where they are convenient for use when lighting up the burners. The push buttons for the pump motors are in turn placed conveniently to the oil pumping, heating and straining equipment which is all contained in a single unit, giving compactness, convenience of manipulation, and ease of observation of thermometers, gages, valves, strainers and regulators.

Ignition on these two burners is effected manually, by inserting a burning kerosene torch into the firebox and then turning on the oil. Once lighted, the burners continue in operation, under a graduated fire in response to the modulation, unless one of the limit or safety controls interferes. Once a burner is so stopped, it will remain off until manually lighted.

A Flexible Boiler Plant

An entirely different method of control was adopted in our third example, the new boiler plant for the Allis-Chalmers tractor plant at La Crosse, Wis. Here there are being installed three 2 pass steel firebox boilers each having 2500 sq ft of heating surface, with provisions for the future installation of a fourth similar boiler. On each boiler there are mounted two rotary oil burners having a combined capacity of 1000 lb of fuel oil per hour, or sufficient to develop approximately 400 boiler hp. The operating boiler pressure will be 40 lb.

It will be noted immediately that this layout gives great flexibility, since the firing range is from one burner up to six. The minimum possible load will always exceed 100 hp, so that one burner can always be kept running. Hence it is merely necessary that the number of burners in operation be kept somewhat in proportion to the load.

View of modern oil fired boilers in large apartment building, illustrating in striking fashion how effectively compactness of arrangement, saving in floor space, and cleanliness are secured

View of boilers at tractor plant showing electrical panel carrying starting switches, CO_2 recorder, and draft indicators. The hydraulic combustion control apparatus is mounted on the steel frame in back of panel

The control problem became one of designing a modulating system that will vary the firing rates of such burners as happen to be running at any momentary load condition. The objective is to maintain steam pressure as nearly constant as feasible with economy in fuel consumption; to accomplish it the following control system was designed.

The first detail is that of oil flow regulation. Various methods are available, embracing apparatus operated electrically, pneumatically and hydraulically. Which of these to utilize can only be decided when other features of the system are considered. For one thing, most valves operate on the principle of varying the admission through an orifice which can be altered by increasing or decreasing the opening with load changes. Such valves are predicated on the existence of a constant oil pressure at the valve inlet. But in this case it was by no means easy to provide for the maintenance of constant oil pressure at all times at all burner inlets. Even though the pressure regulating valves at the pump be capable of keeping the oil pressure constant despite the variations of flow consequent upon load changes, this would not be equivalent to assuring constant pressure at the boiler fronts, for the pressure drop in the oil feed main would in turn be affected by the quantity of oil being burned.

Ordinary pressure reducing valves were considered, for the purpose of leveling off from whatever pressure existed in the feed main to some predetermined pressure to be maintained at each boiler front. From this a further step was taken, to the selection of a special type of pressure reducing valve which can compensate both for the pressure existing momentarily in the feed main and for the load demand. The latter feature is secured by connecting the diaphragm chamber of the reducing valve to a system which adds or detracts an increment of pressure to the upper side of the diaphragm.

By this time it became fairly obvious that, for this particular plant at least, an hydraulic system would be most suitable. This immediately allowed the use of a power cylinder to actuate the damper, the movement of the damper being controlled by attaching the power cylinder in synchronism with the oil regulating valve. As a further refinement, the draft controller was connected through an air flow controller of the differential type, so that it would be responsive to the draft loss through the boiler, which of course is also a function of the load condition.

We now have three special type pressure reducing valves for oil regulation, three damper controllers, and three air flow controllers. To coordinate the operation of all, two more devices are needed. The first is a loading valve, to establish the value of the increment or decrement of oil pressure acting upon the diaphragm of the pressure reducing valves. The second is the master controller itself, otherwise known as the metering control. This is where the connection is made to the steam header, so that any changes in boiler pressure will be immediately reflected in coordinated action of the component elements of the system. The master controller translates changes in steam pressure into impulses which cause the oil flow to vary and the air flow to change in proportion to the fuel being burned.

Another feature of this equipment lies in the hand control valves—one for each air flow controller and one for the master. By turning the hand valve into a selected position, the equipment can be placed under automatic control, or the dampers can be swung instantly to a full open position or a full closed position, as desired. The latter permits closing off dampers of boilers when not being fired. The operating hydraulic medium for the entire system is a light body oil kept under constant circulation by means of a small pumping unit.

The system so far has provided, from the design viewpoint, the factors of steam pressure regulation and efficiency. There remains the element of safety. Here the design reverted to electrical methods. The customary low water cutout, high pressure cutout, flame failure cutout or stack switch, were of course employed in the usual manner. Still another safety control was added—an instrument on each boiler called an excess stack temperature cutout. This is in a sense an auxiliary low water cutout, and acts to stop the burners should the stack gases attain a temperature significantly higher than what should customarily exist, on the theory that such excess might be resulting from a dry boiler.

New Edition of Electrical Handbook

["National Electrical Code Handbook", by Arthur L. Abbott. 4th ed. 1937. 561 + xiii pp. 4½ x 7½ in., semi-flexible binding. Published by McGraw-Hill Book Co., Inc., 330 W. 42nd St., New York, N. Y. Price, $3.00.]

The new edition of this standard handbook is based on the 1937 edition of the National Electrical Code. Its purpose is to enable the reader to grasp readily the general plan, scope, and intent of the code requirements, to present more or less detailed discussions of the rules, and to make the practical application of the rules clear and easily understood.

The book is divided into six major parts, and further is divided into 37 specific sections. The major divisions comprise definitions and general provisions, types of wiring approved under given conditions, installation of materials and apparatus, general requirements for wiring installations, special cases, and construction materials, devices, fittings and equipment.

New Data for

Refrigeration Piping

Carbon Dioxide Liquid and Suction Lines

By R. C. Doremus*

LIQUID carbon dioxide piping is usually extra heavy steel because of the pressure, which is usually 900 to 1050 lb per sq in. according to the condensing water temperature. The liquid has a density very close to water or liquid methyl chloride and is, therefore, heavier than ammonia and lighter than "Freon-12."

As the total heat of CO_2 vapor is relatively low (approximately 138 Btu per lb) and the heat of the liquid is relatively high (approximately 95 Btu per lb) it is evident that the refrigerating effect is very low, or about 43 Btu per lb of liquid circulated. This means that it requires a fluid flow of 4.56 lb per minute of liquid or a little over ½ gpm per ton of refrigeration. This is a rather large flow and it therefore requires care in planning sufficient carrying capacity in the liquid piping. The liquid lines should be proportioned for a fluid velocity in small pipes of not over 4 to 5 ft per second and in larger pipes from 5 to 7 ft per second. Table 1 shows capacities of CO_2 liquid lines used in common practice.

Table 1—Tonnage Capacities for CO_2 Liquid Lines

Size XH Pipe, In.	Tons
½	0–8
¾	9–19
1	20–32
1¼	33–49
1½	50–99
2	100–149
2½	150–224
3	225–299
3½	300–399
4	400–599
5	600–899
6	900–1,250

Obviously, with this rate of flow, the design of CO_2 liquid line piping becomes a hydraulic problem and it is necessary to observe the essentials of hydraulic design. The lines should be as short, straight and direct as possible with no restrictions and a minimum of offsets and fittings. For this reason, as well as the pressure, very few fittings are used and elbows are made of long sweep pipe bends welded to straight lengths and tees are generally more like welded laterals wherever space and arrangement permit this construction in order to favor stream flow principles.

If a plant layout is rather expansive, requiring long runs of pipe, no straight runs should be installed over 100 ft long without making some provision for expansion and contraction of the piping with temperature changes for they are bound to occur, and any expansion or shrinkage should take place where it is planned rather than at an undesired point where it may cause a fracture. Consequently, with very long lines, some changes in direction are necessarily made to permit movement and relieve this expansion and contraction for these pipe lines are welded and, therefore, quite stiff.

Flash Gas in Risers

If the plant is one with very high risers in the carbon dioxide liquid and suction mains, it must be remembered that as the liquid rises from floor to floor to feed evaporators on upper levels, the decrease in static head reduces the internal liquid pressure. If this liquid was condensed in a condenser in the basement from saturated discharge vapor as it must have been, it was stable at the condensing point and any subsequent decrease in pressure in the liquid main without a similar decrease in temperature to the equivalent saturation point, causes the liquid to become unstable and consequently, a portion of the liquid boils off to a vapor until this refrigerating effect reduces the liquid temperature to a stable point for the pressure. This is "flash gas" and is similar to that formed in the low pressure side of the expansion valve due to the drop in pressure from high to low side. This refrigerating effect in the liquid riser frequently chills the pipe line below the dew point of the surrounding air and causes the liquid line to sweat; anti-sweat pipe covering is required to prevent water dripping from the line.

This means that the expansion valve for an evaporator situated very high above the condenser must handle both liquid and gas and will probably require a larger size valve than would be needed to handle liquid only. To keep the velocity within reasonable limits, it may be necessary to increase the size of the piping. This is a point common to all refrigerants but is accentuated in the case of carbon dioxide because the amount of liquid to be fed for any tonnage is a larger quantity than with other refrigerants due to its lower refrigerating effect.

There are very little data or information available on pressure drop of liquid CO_2 in pipe, bends, valves, fittings, etc., so no curves (such as have been given for other refrigerants) are presented for use in designing CO_2 liquid lines on the principle of pressure drop. However, if the designer will be governed by the approximate capacities given, the plant will operate satisfactorily.

In general, the liquid line will be a pipe size one-half the nominal size of the suction line. If the job in ques-

Basic data on the design and installation of piping for industrial and commercial refrigeration are presented in this series of articles. As a refrigeration engineer, Mr. Doremus has had the opportunity to observe jobs which have been poorly designed, has concluded that lack of information is the cause. He therefore makes his data available in this form

*Chief Engineer, Detroit Ice Machine Co. Member of Board of Consulting and Contributing Editors.

For data on piping other refrigerants, see the following issues of Heating, Piping and Air Conditioning: Ammonia Liquid Line Piping, December, 1937; Ammonia Suction Line Piping, January, 1938; "Freon" Liquid Line Piping, August, 1937; "Freon" Suction Line Piping, September, 1937; Methyl Chloride Liquid Line Piping, October, 1937; Methyl Chloride Suction Line Piping, November, 1937.

tion requires a 6 in. suction line, the liquid line will be 3 in. This is a handy rule of thumb to remember for use where pipe size tables are not available.

CO_2 Suction Vapor Characteristics

Not only is CO_2 liquid extremely heavy, but when that liquid boils in the evaporator in producing refrigeration, the resulting CO_2 suction vapor is also extremely heavy. Naturally the liquid requires less space than the vapor for the vapor is the result of the expansion and boiling of the liquid at a lower pressure. In fact, each pound of liquid CO_2 requires a volume of 46 cu in. at 86 F and this same pound of vapor at 5 F requires 460 cu in. or about ten times as much.

The suction vapor of carbon dioxide is heavier than the vapors of other commonly used commercial or industrial refrigerants. When it is realized that the gas is heavy and that more weight of CO_2 is required to produce a ton of refrigeration than with other refrigerants, one can realize that it is a real problem properly to size and plan the arrangement of CO_2 suction lines.

Suction Pipe Line Sizes

In previous articles on suction pipe line sizes for other refrigerants, emphasis has been laid on planning the size of suction pipe lines based on gas velocity and pressure drop, so that the compressor may operate at a suction pressure very near the evaporator pressure with a practical minimum pressure drop in the pipe line and a maximum of compressor volumetric efficiency.

While we do not have much available data on pressure drop in CO_2 lines, we do know that the pressure drop with any refrigerant varies directly as the density of the vapor; this indicates that the proper sizing of the CO_2 suction pipe lines is of paramount importance. Inasmuch as the pressure drop between two points varies directly as the square of the velocity, it is at once apparent that carelessness in the choice of pipe sizes for carbon dioxide is apt to cause even a more detrimental effect than the same action with other refrigerants. A gas velocity that would be satisfactory with other refrigerants will be absolutely ruinous with CO_2.

In order to make it comparatively easy for the designing engineer properly to plan and specify suction pipe line sizes, the writer presents Table 2, which shows unit velocities per ton for CO_2 suction vapor at the various back pressures together with a recommended maximum velocity to be permitted for each pipe size—which of course gives a tonnage capacity for each condition. This has been prepared similarly to the gas velocity charts given in previous articles on other refrigerant vapors and is to be used in the same manner. It will be noted by the reader that if he chooses to permit a higher or lower maximum gas velocity than that indicated by the au-

Three 20 ton CO_2 compressors in hospital cold storage service

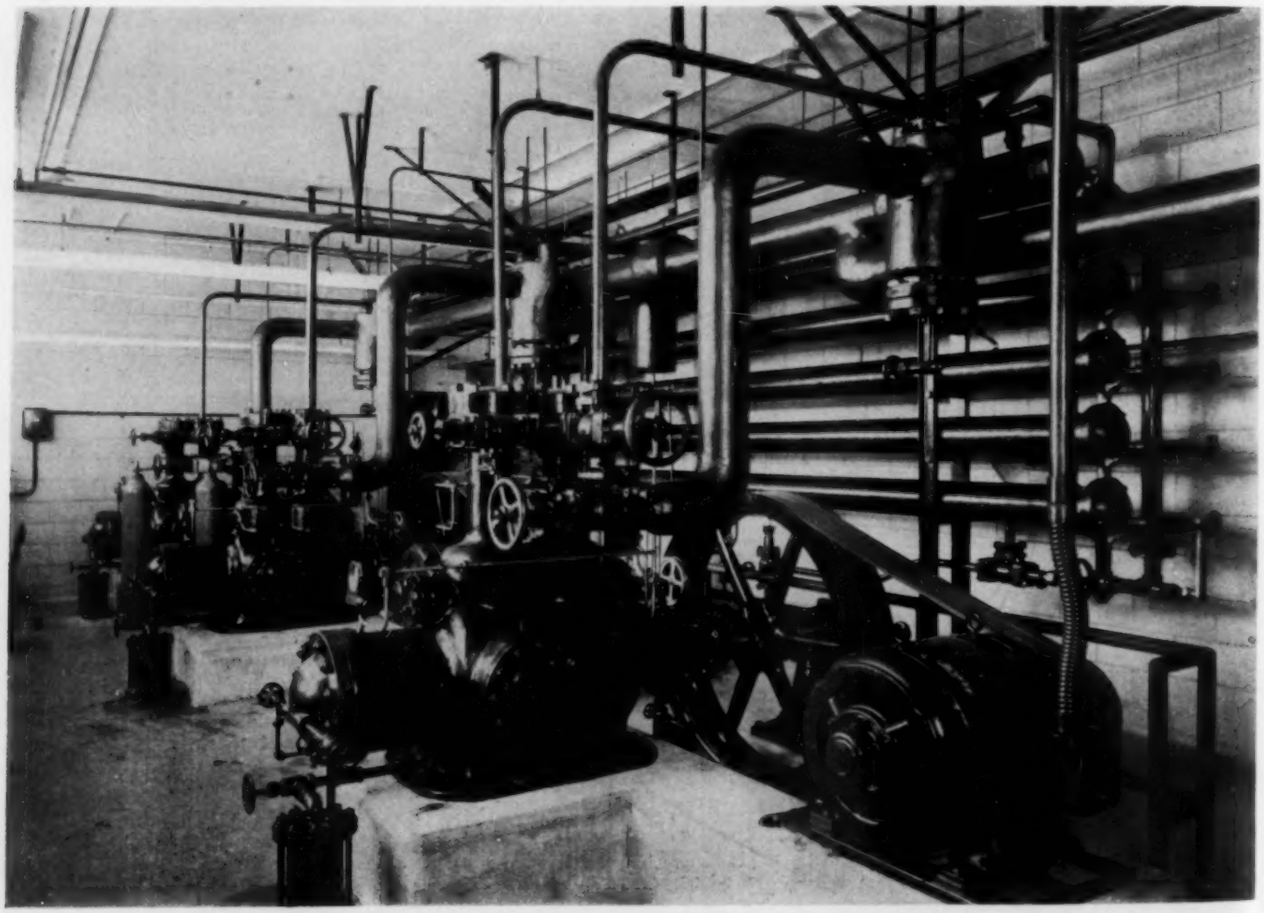


Table 2—Velocity Chart for Carbon Dioxide Suction Vapor at Various Temperatures for Extra Heavy Steel Pipe of Different Diameters, in Feet per Minute per Ton of Refrigeration

Temp. Saturated F	Corresponding Pressure Lb per Sq In.		Sp. Vol. of Vapor Cu Ft per Lb	Total* Heat of Vapor Btu per Lb	Heat of Liquid @ 86 F Btu per Lb	Refrig. Effect Btu per Lb	Weight Lb per Min per Ton	Actual Displacement Cu In. per Min per Ton	Suction Pipe Sizes, In.											
	Abs.	Gage							½	¾	1	1¼	1½	2	2½	3	3½	4	5	6	8	10	12
—40	145.8	131.1	0.6113	137.8	95.0	42.8	4.67	4925	1750	946	570	331	232	139	97.0	62.0	46.2	35.7	22.6	15.7	8.80	5.50	3.80
—35	161.5	146.8	0.5541	138.1	95.0	43.1	4.64	4440	1580	854	515	298	209	125	87.4	56.1	41.6	32.2	20.4	14.2	7.88	4.96	3.42
—30	177.8	163.1	0.5029	138.2	95.0	43.2	4.62	4020	1430	774	467	270	189	114	79.3	50.7	37.7	29.2	18.5	12.9	7.20	4.50	3.10
—25	195.7	181.0	0.4574	138.4	95.0	43.4	4.61	3645	1300	700	422	245	172	103	71.8	46.0	34.2	26.5	16.7	11.6	6.52	4.08	2.80
—20	214.9	200.2	0.4168	138.5	95.0	43.5	4.60	3315	1180	637	384	222	155	93.5	65.2	44.7	31.1	24.0	15.2	10.6	5.93	3.70	2.55
—15	235.4	220.7	0.3769	138.7	95.0	43.7	4.57	2978	1162	572	345	200	140	84.0	58.3	37.5	27.0	21.6	13.6	9.51	5.15	3.35	2.29
—10	257.3	242.6	0.3472	138.7	95.0	43.7	4.57	2745	976	528	318	184	129	77.6	48.2	34.6	25.8	19.9	12.6	8.82	4.91	3.07	2.11
—5	280.6	265.9	0.3175	138.8	95.0	43.8	4.56	2510	894	482	291	168	118	71.0	45.3	31.7	23.5	18.2	11.5	8.03	4.50	2.80	1.93
0	305.5	290.8	0.2904	138.9	95.0	43.9	4.55	2290	815	440	266	154	108	64.8	43.3	28.9	21.5	16.6	10.5	7.42	4.10	2.56	1.76
5	332.0	317.3	0.2660	138.8	95.0	43.8	4.56	2090	744	402	242	140	98.6	57.0	41.2	26.4	19.6	15.2	9.61	6.54	3.61	2.33	1.62
10	360.2	345.5	0.2437	138.7	95.0	43.7	4.57	1927	686	370	224	129	91.1	54.5	38.0	24.3	18.1	14.0	8.83	6.11	3.44	2.15	1.48
15	390.9	375.3	0.2236	138.5	95.0	43.5	4.60	1785	665	344	207	116	84.3	50.5	35.2	22.4	16.8	12.3	8.82	5.42	3.04	1.90	1.31
20	424.8	407.1	0.2049	138.3	95.0	43.3	4.62	1635	582	314	188	110	77.2	46.2	32.1	20.6	15.3	11.8	7.53	5.24	2.91	1.82	1.26
25	463.3	440.6	0.1879	138.0	95.0	43.0	4.65	1508	537	290	175	97.7	71.2	42.5	29.7	19.0	14.1	10.9	6.94	4.82	2.70	1.68	1.16
30	490.8	476.1	0.1722	137.8	95.0	42.8	4.67	1392	496	268	161	90.4	66.3	39.4	27.5	17.5	13.1	10.1	6.41	4.43	2.49	1.56	1.07
35	529.4	514.7	0.1576	137.2	95.0	42.2	4.74	1293	461	249	150	84.0	61.1	36.6	25.5	16.3	12.1	9.41	5.92	4.13	2.31	1.44	1.00
40	567.8	553.1	0.1444	136.7	95.0	41.7	4.80	1200	427	231	139	77.8	56.6	33.9	23.6	15.1	11.3	8.72	5.50	3.84	2.15	1.34	0.92
41	571.1	556.4	0.1419	136.5	95.0	41.5	4.82	1180	420	227	137	76.5	55.8	33.3	23.2	14.9	11.1	8.64	5.42	3.80	2.11	1.32	0.91
Maximum velocity recommended for tonnage capacity in any pipe size to avoid excessive pressure drop in suction mains									1100	1300	1500	1550	1600	1650	1700	1750	1800	1850	1900	1950	2000	2200	2400

Example: What size CO_2 suction line should be used for a load of 100 tons at 15 F?
If we consider a 3 in. pipe, $100 \times 22.4 = 2240$ fpm. This is too high a velocity, as the limit for 3 in. pipe is 1750 fpm.
If we consider a 3½ in. pipe, the velocity will be $100 \times 16.8 = 1680$. This is satisfactory, inasmuch as the limit velocity for 3½ in. is 1800 fpm.

thor, he is at liberty to do so on account of the flexible nature of the tabulated data.

This table includes data on 5 F intervals from —40 F to +41 F. It shows the actual compressor displacement required varies from 4925 to 1180 cu in. per minute per ton. The theoretical compressor displacement will be more than these quantities inasmuch as no compressor operates with 100 per cent volumetric efficiency and the compressor displacement must therefore be modified to take care of this volumetric efficiency loss.

The table shows the actual gas velocities in each size suction pipe at these various evaporator temperatures expressed in feet per minute per ton of refrigeration. Naturally, it follows that if we know the total refrigerating capacity involved, we need only multiply the total tons to be conveyed through the pipe by the unit velocity of one ton, to obtain the resultant gas velocity for any load through any size pipe at any evaporator temperature. This of course is based on handling saturated vapor, whereas the true condition in any plant will involve a slight amount of vapor superheat for safe compressor operation; this will tend to increase the velocity by a small percentage according to the superheat.

Examples Show Use of Data

To illustrate the use of these data, assume we are considering a carbon dioxide refrigerating plant in which ice is manufactured in a large department store. Let us presume the production is 30 tons a day and that on account of losses the plant must have a 50 ton refrigerating capacity at 0 F evaporator temperature for a 15 F brine temperature.

In a 2½ in. suction line: $50 \times 43.3 = 2165$ fpm. This is too high a velocity for 2½ in. which has a limit of 1700 fpm.

In a 3 in. suction line: $50 \times 28.9 = 1445$ fpm. This is satisfactory for 3 in. which has a limit of 1750 fpm.

Now let us suppose we have the same 50 ton load in another plant operating at an evaporator temperature of —40 F and 131.1 lb gage suction pressure. Will the same 3 in. suction pipe line be sufficient? If not, what size should it be?

For 3 in. pipe, $50 \times 62 = 3100$ fpm. This is too high for a 3 in. pipe where the limit is 1750 fpm.

For 3½ in. pipe, $50 \times 46.2 = 2310$ fpm. This is even too high for a 3½ in. pipe where the limit is 1800 fpm.

For 4 in. pipe, $50 \times 35.7 = 1785$ fpm. This is O.K. for a 4 in. pipe where the limit is 1850 fpm.

Let us suppose we have the same 50 ton load operating in an air conditioning plant in a small theater at an evaporator temperature of +41 F and a suction gage pressure of 556.4 lb per sq in. How does this affect the size of suction pipe line? What size pipe should be used?

For 3 in. pipe, $50 \times 14.9 = 745$ fpm which is a very low vapor velocity.

For 2½ in. pipe, $50 \times 23.2 = 1160$ fpm which is still quite low in velocity.

For 2 in. pipe, $50 \times 33.3 = 1665$ which is approximately the limit for a 2 in. pipe and this is therefore O.K.

Thus the same 50 ton load requires a

2 in. suction pipe at an evaporator temperature of + 41 F;
3 in. suction pipe at an evaporator temperature of + 15 F;
4 in. suction pipe at an evaporator temperature of — 40 F;

regardless of what suction connection size may be fur-

[Concluded on page 124]

Heating, Piping and Air Conditioning's

Piping Review

IT is the object of this review to place pertinent data on recent piping installations on the record for general and convenient reference. This month's table should be regarded merely as the first step, for similar information on other projects will be published as it becomes available. Suggestions for making this review of the maximum usefulness will be appreciated by the editors.

The cooperation of the engineers who have furnished the data presented this month is gratefully acknowledged.

HEATING, PIPING AND AIR CONDITIONING'S Piping Review

NAME AND LOCATION — ENGINEERS AND CONTRACTORS	PIPING SERVICE	PRESS. AND TEMP. LB PER SQ IN. AND F	NOM. DIAM. AND WALL THICKNESS IN.	COMPOSITION OF PIPE MATERIAL	FLANGED, ELECTRIC OR GAS WELDED?	STRESS RELIEVED?	JOINTS X-RAYED?	KIND OF GASKETS	VALVE MATERIAL: BODY	VALVE MATERIAL: TRIM
Conners Creek, Detroit, Mich. Engineers, The Detroit Edison Co. Piping, Construction Bureau, The Detroit Edison Co.	Superheated steam	650 lb, 850 F superheater outlet	4 to 16. Sched. 80	Silicon "killed," Grade B, A106	Electric welded[a]	Yes	No	Asbestos composition[a]	Cr-Ni-Mo	Stellite in later sections. Cu-Ni in first sections
	Boiler feed	900 lb, 375 F	6 and 8. Sched. 80	Silicon "killed," Grade B, A106	Flanged and electric Welded	Yes	No	Asbestos composition	Carbon steel C-Mo	Cu-Ni alloy
	Saturated steam	710 lb, 507 F	2, 2½, 3. Sched. 80	Medium carbon, Grade B, A106	Flanged and Electric welded	Yes	No	Asbestos composition	Carbon steel	Cu-Ni alloy
	Saturated steam	250 lb, 410 F	¾ – 4. Sched. 40	Grade A, A106, or B	Electric welded, screwed flanged	Yes	No	Asbestos composition	Carbon steel	Cu-Ni alloy
Delray, Detroit, Mich. Engineers, The Detroit Edison Co. Piping, Construction Bureau, The Detroit Edison Co.	Superheated steam	865 lb, 910 F superheater outlet	4 to 14 in.[b] Sched. 100	C-½% Mo	Electric welded[c]	Yes	No	Probably furniture iron[c]	Cr-Ni-Mo C-Mo	No. 6 stellite
	Boiler feed	1200 lb, 400 F	6 and 8. Sched. 80	Silicon "killed," Grade B, A106	Electric and flanged	Yes	No	Asbestos composition	Carbon steel	No. 6 stellite
	Saturated steam	950 lb, 540 F	2, 2½, 3. Sched. 80	Medium carbon, Grade B, A106	Electric and flanged	Yes	No	Asbestos composition	Carbon steel	Cu-Ni alloy
	Saturated steam	200 lb, 390 F	¾ – 4. Sched. 40	Grade A or B, A106	Elec. welded, screwed, flanged	Yes	No	Asbestos composition	Carbon steel	Cu-Ni alloy
Market Street Station, New Orleans, La., 1938 30,000 Kw Extension Consulting Engineers, Ebasco Services, Inc. Piping Contractors, National Valve & Mfg. Co.	High pressure steam	875 lb, 905 F	12 0.843	Carbon-molybdenum	Electric welded[d]	Yes	No	Monel metal, dead soft	Cast carbon-moly	Stainless steel, stellited disc and seat rings
	Auxiliary steam	360 lb, 700 F	6 0.432	Carbon steel	Electric welded	Yes	No	1/16 in. Durabla	Cast carbon	Stainless steel, monel metal
	Boiler feedwater	1230 lb, 385 F	8 0.500	Carbon steel	Electric welded[d]	Yes	No	Monel metal, dead soft	Cast carbon	Stainless steel, stellited disc and seat rings
	Condensate	150 lb, 210 F	8 0.322	Carbon steel	Electric welded	No	No	1/16 in. rubber	Cast carbon	Stainless steel
Mountain Creek Station, Dallas, Texas, 1938 31,250 Kw Construction Consulting Engineers, Ebasco Services, Inc. Piping Contractors, not awarded at time of report	High pressure steam	650 lb, 825 F	12 0.843	Carbon-molybdenum steel	Electric welded[e]	Yes	No	Monel metal, dead soft	Cast carbon-moly	Stainless steel, stellited disc and seat rings
	High pressure boiler feed	900 lb, 240 to 375 F	8 0.500	Carbon steel	Electric welded[e]	Yes	No	Monel metal, dead soft	Cast carbon	Stainless steel, stellited disc and seat rings
Riverside Station, Shanghai, China, 1938 15,000 Kw Extension	Main steam	1220 lb, 920 F	12¾ 1.312	Carbon-molybdenum	Electric welded[f]	Yes	No	None	Carbon-moly	Stainless steel
Consulting Engineers, Ebasco Services, Inc. Piping Contractors, not awarded at time of report	Boiler feed	1700 lb, 230 to 380 F	8¾ 0.812	Carbon	Electric welded[f]	Yes	No	None	Carbon-moly	Stainless steel

[a]Connections to superheater safety valves and to throttles of three 20,000 kw units and to the first three boilers were made with flanged joints. Balance of main steam joints are welded. Connections to house service turbines and to emergency boiler feed pump turbines are flanged. All flanged joints in 850 F service use asbestos composition gaskets.

[b]4 and 6 in. pipe corresponds to Schedule 120, since Schedule 100 does not extend to the smaller sizes.

[c]Connections to superheated steam safety valves and to a 10,000 kw topping unit are the only flanged joints involved in main steam. Some other type of metal gasket may be substituted for furniture iron if later found desirable. In auxiliary superheated steam, connections to house service turbines and to emergency boiler feed pump turbines are flanged. None of this piping has been installed at date reported, 12/1/37.

[d]Joints for high pressure steam and water made with recessed chill rings and upset pipe ends.

[e]Joints for high pressure steam and boiler feedwater made with recessed chill rings; flanges at boiler superheater outlet and turbine inlet.

[f]Main steam and boiler feed piping joints welded, including those between valves and piping, except that flanged connections used at superheater outlets, economizer outlets and feed pump discharge nozzles.

Ford's Standards for Air Conditioning

Twin City Branch Assembly Plant an Example of Their Use

By J. J. Floreth*

STANDARDS for air conditioning established by the Ford Motor Co. were discussed in the December HEATING, PIPING AND AIR CONDITIONING.[1] It is believed that Ford's program of and investment in air conditioning is the largest under single ownership in modern industry, and Ford installations are serving manufacturing processes and human comfort in buildings throughout the world.

Typical of the installations being made in their assembly plants throughout the country is the one completed last July at the Twin City branch, St. Paul, Minn. The plant itself is spread over 20 acres, and situated on the banks of the Mississippi river, above the city of St. Paul.

Ford invited five leading firms to submit bids on a complete air conditioning system to serve the general and private office areas, showroom, dealers' room, service school room, first aid rooms, and a group of seven factory offices situated at the north end of the assembly plant. The people occupancy, light wattage and other load factors shown in Table 1 were established by the specifications and a joint meeting of the bidders with the plant superintendent and a member of the power and construction department. Design temperatures were set at the usual standard of 80 F dry bulb and 45 per cent relative humidity to be maintained inside when the outside temperatures were 95 F dry bulb and 75 F wet bulb, with winter conditions of 70 F and 30 per cent relative humidity when the outside temperature ranged to as low as minus 30 deg, which is not uncommon in that section of the country. The design selected by Ford established a total internal load of approximately 89 tons refrigeration required for summer duty, to which the outside air quantity of 12,250 cfm added 33 tons, resulting in a total system capacity of 122 tons refrigeration. The quantities of outside air supplied to each conditioned space were based on producing approximately two complete outside air changes per hour, which is above the average for office spaces in which smoking is not permitted, but assures the fresh wholesome atmosphere desired. Even this standard is stepped up in the case of first aid rooms, where Ford desires 50 per cent of the air supplied to be outside air, providing over four complete changes per hour in most cases. As a result the first aid rooms maintain the same fresh atmosphere and absence of odor found in the office spaces, in spite of the medicines, antiseptics, and body odors encountered. The unusual height of the ceilings throughout the conditioned spaces assisted the problem of air distribution, and the total quantities of air supplied to each zone were therefore established from the air change required to give adequate movement of the room volumes, preventing stratification.

Although the load of the showroom and lobby was based on a maximum occupancy of 600 people, in order to maintain satisfactory conditions when this peak existed, the infrequency of this loading offered a problem in the design of the systems, if operating economy was to be obtained. The normal occupancy of the show room varies from perhaps six to 50 people, and the maximum load occurs only four or five times a year when the dealers and distributors from the surrounding territory assemble to preview new models or attend some form of policy or sales meeting. The problem was further influenced by the fact that some of these meetings occur at night, when the general and many of the private offices are not in use, at which time the system serving these areas would normally not be in operation. However, other meetings and displays conducted in the showroom are often in progress during the daytime, which made it inadvisable to attempt to handle the showroom peak with the same system that served the offices, by sacrificing proper conditions in the office spaces during those periods.

The location and arrangement of the offices in relation to the showroom, dealers' room, and service school also clearly indicated the construction advantages of installing a separate and independent system to serve the general and private office zone and the design was approached from this standpoint. The service school area is seldom in use during large gatherings in the showroom, and the fact that this space is directly adjacent to the

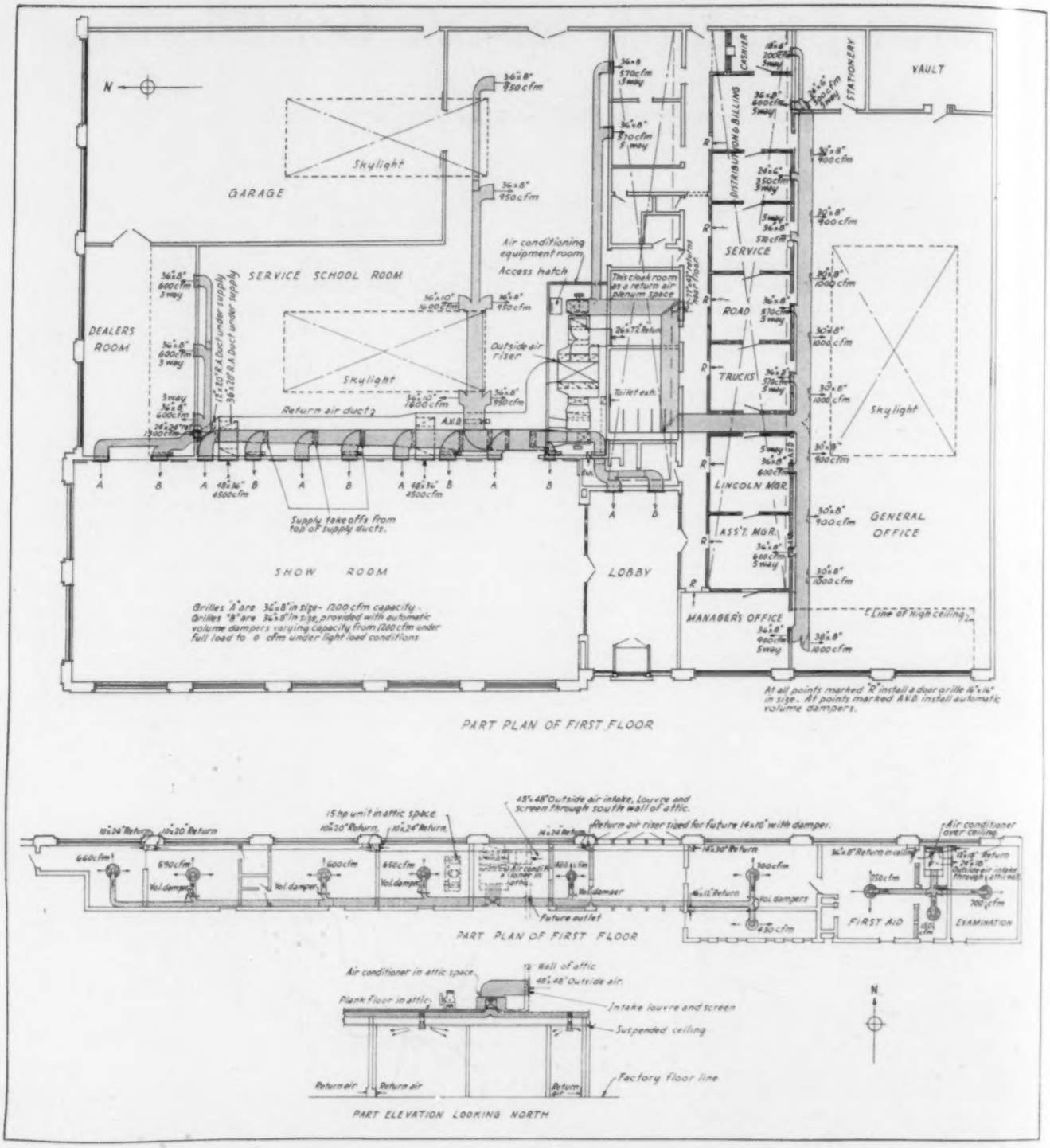
Table 1—Air Conditioning Data for Branch Assembly Plant

	Show Room	Assembly Room	Service School	General Office	Private Offices	Factory Offices	First Aid Rooms	Totals
Floor area, sq ft	5,180	945	4,815	5,200	3,320	1,565	760	21,785
Ceiling height	22'6"	18'0"	22'0"	14'0"	14'0"	14'0"	14'0"	
Volume, cu ft	116,400	17,000	105,900	73,000	46,500	21,900	10,640	391,340
People	a600	75	30	50	28	48	25	856
Light Wattage	5,100	900	1,600	7,200	5,800	2,600	1,400	24,600
Motor hp								11 5
Internal sensible, tons	25 8	3 3	13 5	14 9	10 6	5 8	2 3	76 2
Internal latent, tons	8 8	1 1	0 5	0 8	0 4	0 7	0 4	12 7
Total internal, tons	34 6	4 4	14 0	15 7	11 0	6 5	2 7	88 9
Outside air, cfm	4,000	600	2,000	2,500	1,600	750	800	12,250
Outside air, change per hr.	2 1	2 1	1 1	2 1	2 1	2 0	4 5	
Outside air per person, cfm	6 7	8	67	50	57	15 5	32	
Outside air, tons ref	10 8	1 6	5 4	6 7	4 3	2 0	2 2	33 0
Total sensible, tons	31 0	4 1	16 0	18 4	12 9	6 8	3 4	92 6
Total latent, tons	14 4	1 9	3 4	4 0	2 4	1 7	1 5	29 3
Total tons refrig	45 4	6 0	19 4	22 4	15 3	8 5	4 9	121 9
Total air supplied, cfm	7,200	1,800	7,000	8,600	6,400	4,000	1,600	36,600
Air recirculated, cfm	3,200	1,200	5,000	6,100	4,800	3,250	800	24,350
Air changes per hr	3 7	6 4	4 0	7 0	8 2	10 9	9 0	

[a] Air normally supplied to service school is transferred to showroom when occupancy approaches 600.

*Air Conditioning Engineer, York Ice Machinery Corp.

[1] "Ford Motor Company Sets High Air Conditioning Standards," by J. J. Floreth, HEATING, PIPING AND AIR CONDITIONING, December, 1937, pp. 729-732.


GARAGE
Skylight
SERVICE SCHOOL ROOM
DEALERS ROOM
SHOW ROOM
LOBBY
SERVICE
ROAD
TRUCKS
LINCOLN MGR.
ASS'T. MGR.
MANAGER'S OFFICE
GENERAL OFFICE
VAULT
STATIONERY
CASHIER
DISTRIBUTOR & BILLING
Air conditioning equipment room
Access hatch
This cloak room as a return air penum space
Outside air riser
Return air duct
Toilet exh.
Supply take offs from top of supply ducts.
Grilles 'A' are 36"x8" in size - 1100 cfm capacity.
Grilles 'B' are 36"x8" in size, provided with automatic volume dampers varying capacity from 1100 cfm under full load to 0 cfm under light load conditions
At all points marked 'B' install a door grille 16"x16" in size. At points marked A.V.D. install automatic volume dampers.
PART PLAN OF FIRST FLOOR
15 hp unit in attic space
48"x48" Outside air intake, Louvre and screen through south wall of attic.
Return air riser sized for future 14x10" with damper.
Air conditioner over ceiling
Vol. damper
Air cond'g. in attic
Future outlet
FIRST AID
EXAMINATION
PART PLAN OF FIRST FLOOR
Air conditioner in attic space
South wall of attic
48"x48" Outside air intake louvre and screen
Plank floor in attic
Suspended ceiling
Return air
Factory floor line
PART ELEVATION LOOKING NORTH

showroom and can therefore be served by the same air supply trunk, led to the design eventually adopted. From the data tabulated in Table 1 it will be noted that the total of 19.4 tons refrigeration required by the service school room, practically balances the heat generated by 600 people when taken at an average value of 400 Btu per person. By shutting off the air supply to the service school entirely, this 19.4 tons of refrigeration could be made available for the infrequent showroom peaks, if an arrangement was provided for properly distributing this additional air quantity to the showroom. This was accomplished by the duct design, grille arrangement, and automatic damper control indicated on the plan drawing of the office and showroom systems.

Scheme of Air Distribution

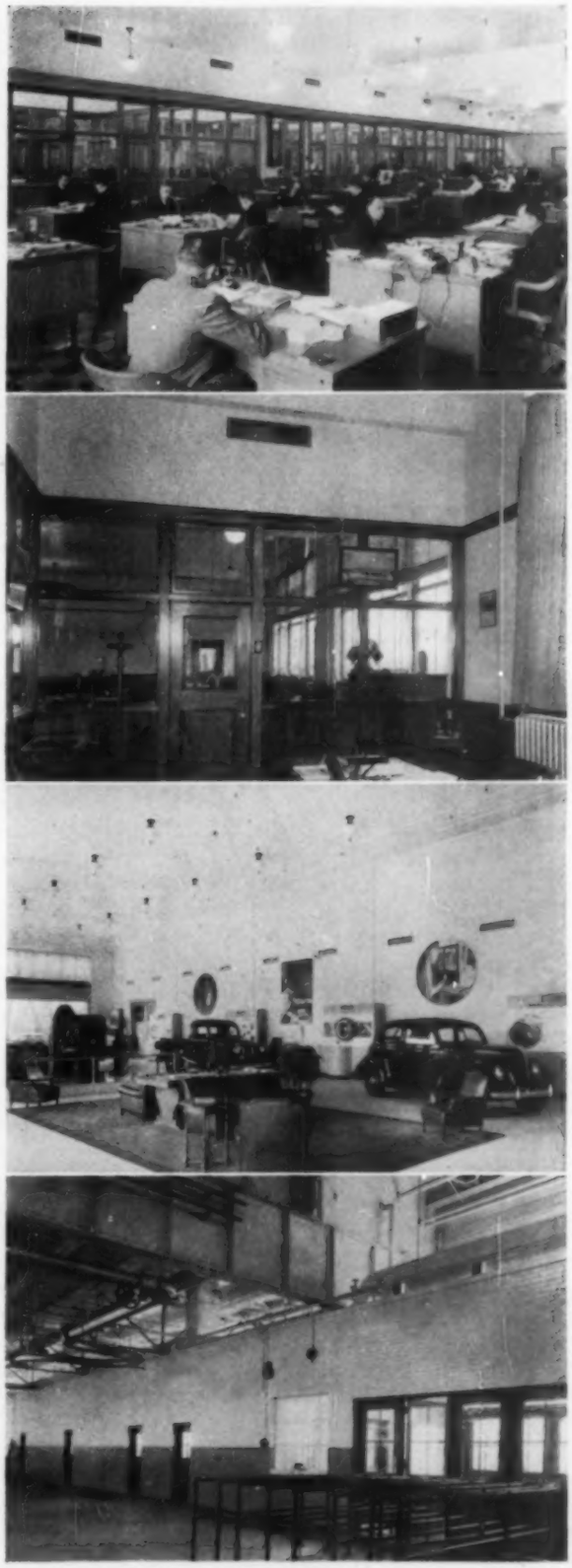
The supply fan serving the showroom, dealers' room and service school discharges into a main air supply trunk situated just under the structural steel supporting the roof, and running through the service school room along the wall separating this area from the showroom.

Twelve outlets are taken directly off this air supply main, and carried through the wall to distribute air evenly the entire length of the showroom and lobby. A branch duct is carried across the north end of the service school to supply three outlets serving the dealers' room and another branch, taken from the supply trunk not far from the fan discharge, is carried from west to east across the width of the service school to supply conditioned air to this area by means of six outlets discharging in opposite directions.

A main volume damper operated by a two position damper motor is situated in the branch duct supplying the service school, and operates in conjunction with two position volume dampers mounted in five of the outlets supplying the showroom. All of these dampers are under the manual control of a single throw switch mounted on the control panel in the compressor room, which permits their positions to be reversed from this common point when the increased occupancy of the showroom calls for the additional refrigerating capacity available in the service school.

During normal daily operation of the system the main damper in the branch supplying the service school is in the full open position, determined by previous setting to supply 7000 cfm of conditioned air to this area, and only 7200 cfm of air is being supplied to the showroom and lobby through seven outlets, to absorb heat generated during normal occupancy. When peak loads occur in the showroom the positions of these dampers are reversed by means of the control switch, shutting off all air supply to the service school and opening the other six outlets in the showroom to permit each of them to distribute approximately 1200 cfm for which they are designed. This not only permits shifting of the refrigerating capacity to accomplish the most economical operation, but also increases the air circulation in the showroom proportionately to prevent objectionable diffusion temperatures, and increases the quantity of outside air supplied to the heavily populated area. These features are accomplished without disrupting the air distribution from the supply outlets, inasmuch as each grille handles 1200 cfm of air, maintaining the same velocity and trajectory, regardless of the total quantity being supplied to the showroom.

The general and private offices at the south end of the building are served by a separate system, the arrangement of which may also be traced from the plan drawing. Each private office is supplied by a separate outlet, with nine grilles serving the general office area. The offices of the plant manager, assistant manager, and Lincoln manager, at the west end of the office group, were provided with individual temperature adjustment by the modulating volume dampers mounted in the supply outlets, and operated by individual thermostats in these offices. Although normally maintained at the same conditions as the rest of the office areas, this feature allows each of the plant's major executives a slight leeway in adjusting temperatures to suit individual preference, or to compensate for the unusual occupancy of meetings and conferences. The design of the office system employs the building corridor as a return air duct, with 45 deg deflecting grilles neatly mounted in the lower panel of each corridor door to permit recirculation of a portion of the air supplied to each office.



These views indicate the care exercised in installing air conditioning equipment. From top to bottom: General office showing furred in ducts and neat grille arrangement. . . . . Plant manager's office with concealed duct work. . . . . The showroom. . . . . Exterior of the factory offices conditioned by the smaller system. The main supply ducts may be seen above the offices

Service school room, showing in the background the glass panelled air conditioning machine room, and ducts

Grilles Blend With Architectural Treatment

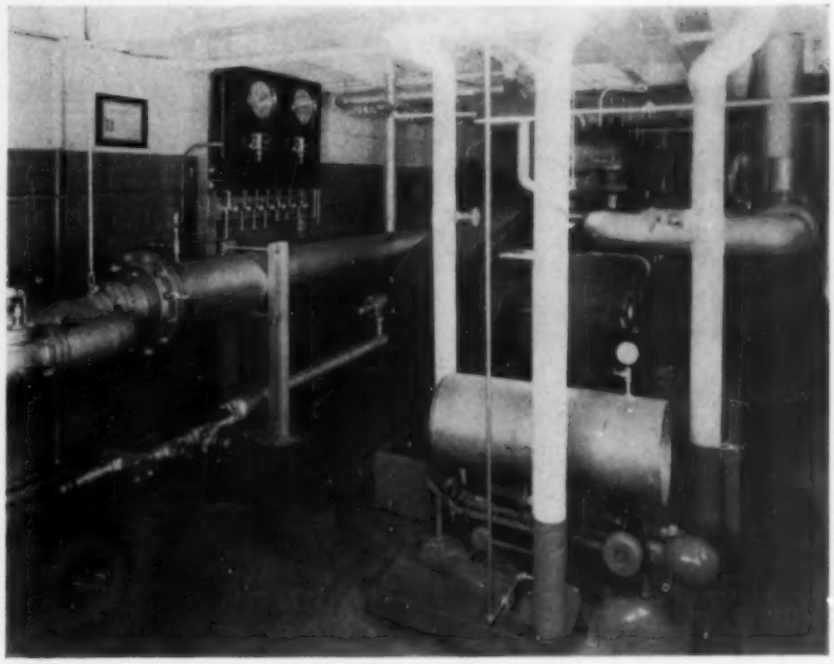
Interior views of the showroom, general office, and manager's office indicate the finished appearance and architectural treatment of the installation. Inasmuch as the service school is essentially a work space, with exposed steel truss and roof members, the duct work passing through this room was not furred in, but was given two coats of white enamel to harmonize with the semi-gloss of the white painted walls. The lower part of these walls is painted a light gray to a height of approximately 5 ft, trimmed with a bright red band, and where the return ducts from the showroom and dealers' room drop to the floor line in the service school they are painted and striped in a similar manner to present an overall appearance pleasing to the eye. The only parts of the air conditioning system visible in the showroom and dealers room are the 36 in. by 8 in. directional flow grilles in the east and south walls. The original design contemplated putting these grilles at a height of approximately 20 ft in order to bring them in line with the top of the wall panels, but a structural steel roof truss carried in the upper part of this wall necessitated lowering the grilles to a height of 12 ft in order to clear this obstruction. The grilles were furnished in prime finish and carefully painted to match exactly the wall color.

All ducts passing through the office area are furred in, with grilles mounted flush with the plaster line, to resemble a large beam in the general office. Antique bronze finish grilles are used throughout, including return grilles in corridor doors, and blend beautifully with the walnut woodwork and furniture. Where the main air supply duct runs through the corridor to feed the general office trunk, the corridor ceiling was lowered to a height of 9 ft, level with the top of the wood wall panels, and completely conceals this transverse run. Only a grille mounted flush with the plaster wall in any private office is visual evidence of the air conditioning.

Visitors See Machine Room

The entire air conditioning and refrigerating equipment for the two systems serving the office and showroom areas is situated in an enclosed machinery room at one corner of the service school. Due to the 22 ft, 6 in. ceiling height available at this point it was possible to conserve floor space by placing the equipment on two levels, with compressors, condensers, motors and similar heavy apparatus mounted on suitable cork isolated concrete foundations at the building floor line and fans, cooling and heating coils, filters and accessory air handling equipment supported at a height of 8 ft by a sturdy platform constructed of 2 in. tongue and groove oak planking, supported and reinforced by steel I beams. After all equipment had been erected and preliminary adjustments completed, a 4 in. tile and plaster partition was erected to enclose completely the apparatus; it is provided with a steel door and access ladder to the upper level. To improve the appearance of this enclosure and permit visitors to observe the operation of the refrigerating and air conditioning equipment, two large steel sash windows were set in the long side wall, and reveal the brilliantly lighted interiors where the refrigerating equipment, painted aluminum bronze with red trim, and the white enamel finish of the air conditioning equipment form a striking machine room.

The two separate air systems have a common outside air intake plenum which is extended through the roof directly overhead, and terminates in a sheet steel pent house, provided with suitable rain louvers and screen, sized to permit 100 per cent outside air to be introduced during those seasons of the year when outside temperatures allow complete ventilation to maintain satisfactory conditions without the assistance of the heating or refrigerating equipment. From this plenum the outside and recirculated air mixture of each system is drawn through the filters, finned cooling and heating coil surface, and moisture eliminators of the air conditioning units. The refrigeration cycle is a direct expansion system, using "Freon-12" refrigerant, and employing two 8 in. bore by 6 in. stroke two cylinder, vertical, single acting, enclosed type compressors designed especially for use with this refrigerant. These compressors are cross connected, with the suction line from each bank of cooling coils brought together into a common header from which each machine takes its individual suction connection, and discharging into a single 12 in. diameter and 15 ft long horizontal shell and finned tube type "Freon-12" condenser.



Ground floor of air conditioning machine room, emphasizing the finished appearance of the installation

Capacity and Bypass Control

Each compressor is equipped with 50 per cent automatic capacity reduction devices, and is V-belt driven by a 50 hp, 1200 rpm automatic start motor of the special Ford design, which results in four completely automatic steps of refrigeration capacity control, namely 100, 75, 50 and 25 per cent of full load. This capacity flexibility was considered essential to provide maximum operating economy, in view of the large variation possible in total load due to showroom and dealer room occupancy, sun effect on the exposed roof, west wall and glass surfaces, and the wide range of outside temperatures experienced in that locality during the summer months. The wisdom of this design feature is borne out by operating logs obtained during the past season, which indicate that the plant operated at a load factor between 50 and 75 per cent the majority of the time.

Each system uses the temperature control method in which automatic dampers across the face area of the cooling coils and in the bypass connection around these coils regulate the proportions of air conditioned by contact with the coil surface, and that bypassed directly to the fans. Modulating thermostats in each zone control the positioning of these dampers, and are compensated by a master thermostat in the outside air stream, so as to maintain a sliding differential with the outside temperature, ranging from an inside temperature of 80 F when the outside is 95 F to an inside temperature of 70 F when the outside temperature is 70 F. This compensating control feature is standard on all Ford human comfort installations, as Ford engineers are fully aware of the danger to health in holding unusually low inside conditions in order to produce a constant differential as the outside temperature drops off, which is unfortunately still the practice of many installations where the advertising value of an appreciable temperature reduction is given primary consideration. The operation of compressors and corresponding control of refrigerating capacity is accomplished by additional modulating thermostats and humidistats in the separate return air streams from each conditioned zone, and interconnected with position switches on the coil face dampers, so that both components of the design condition—temperature and relative humidity—must be satisfied.

The dry bulb temperature during the heating season is controlled by low limit thermostats mounted in the discharge of the supply fans, acting in conjunction with high limit thermostats in the separate return air streams, which position modulating valves controlling the steam supply to the heating coils. This arrangement prevents either objectionable drafts occasioned by supplying air at too low a temperature, or over-heating the conditioned spaces when operating in conjunction with the manually controlled direct radiation. Mist type atomizing nozzles in each unit are under the control of winter humidistats, which are manually set to the desired per cent of relative humidity. The operation of all summer and winter instruments is under the single control of a summer-winter switch mounted on the control panel in the compressor room, from which point they can be thrown out of operation entirely to permit manual operation of the systems during tests and service adjustments, or placed in complete position for summer or winter operation by merely throwing this single switch. These various automatic temperature controls are, of course, supplemented by complete safety devices and controls.

Units for Factory Offices

Although smaller in every respect and of only approximately 13 tons total refrigerating capacity, the separate air conditioning systems serving the group of seven factory offices and four first aid rooms at the north end of the factory building are similar in every way to the larger units handling the general office and showroom zones. These offices have an interior height of 14 ft, 0 in. but the roof height of the factory building itself, approximately 30 ft, allows considerable headroom above them, a portion of which is enclosed to form an attic space. In order to conserve the more valuable factory floor area the refrigerating and air conditioning units for these systems were situated in this attic, with all piping connections close coupled and the air distributing duct work run on top of the office ceilings below, as indicated by the plan drawing. This arrangement made it advisable to use ceiling plaques for the distribution of the conditioned air, drawing the air to be recirculated from the floor of each room by means of small ducts (one for each two offices) run down in the corners.

High Temperature Piping Joints

A report of the American Society of Mechanical Engineer's recent meeting on high temperature joints, which included discussions of an investigation of stress conditions in a full size welded branch connection, a test to destruction of the manifold, an analysis of bolted joints at high temperatures, stresses and deformations in pipe flanges subjected to creep at high temperatures, and the effect of temperature variations on creep strength of steels

PRINCIPLES of joint making, applied to both welded and mechanically joined pipe lines, were analyzed and emphasized in a series of excellent papers read at the December 8 session of the annual meeting of the American Society of Mechanical Engineers in New York City.

Stress Conditions in Welded Branch Connection

Results of an "Investigation of Stress Conditions in a Full Size Welded Branch Connection" were contributed by F. L. Everett, assistant professor of engineering mechanics, University of Michigan, and Arthur McCutchan, engineer, engineering division, The Detroit Edison Co. "The purpose of the investigation was to determine: (a) The magnitude and location of the stresses in and adjacent to the welded juncture of branch and header; (b) the effect of applying a reinforcing ring; and (c) the ability of the ring reinforced branch connection to develop the full strength of the pipe."

The header, which was constructed by The Detroit Edison Co., is shown in Fig. 1, which includes details of the three types of end closure used; Fig. 2 shows the ring type reinforcement, similar to that recommended by the boiler code for reinforced nozzle openings. The manifold was stress relieved by heating to 1150 F before the initial tests and again after the reinforcing ring had been fused into place.

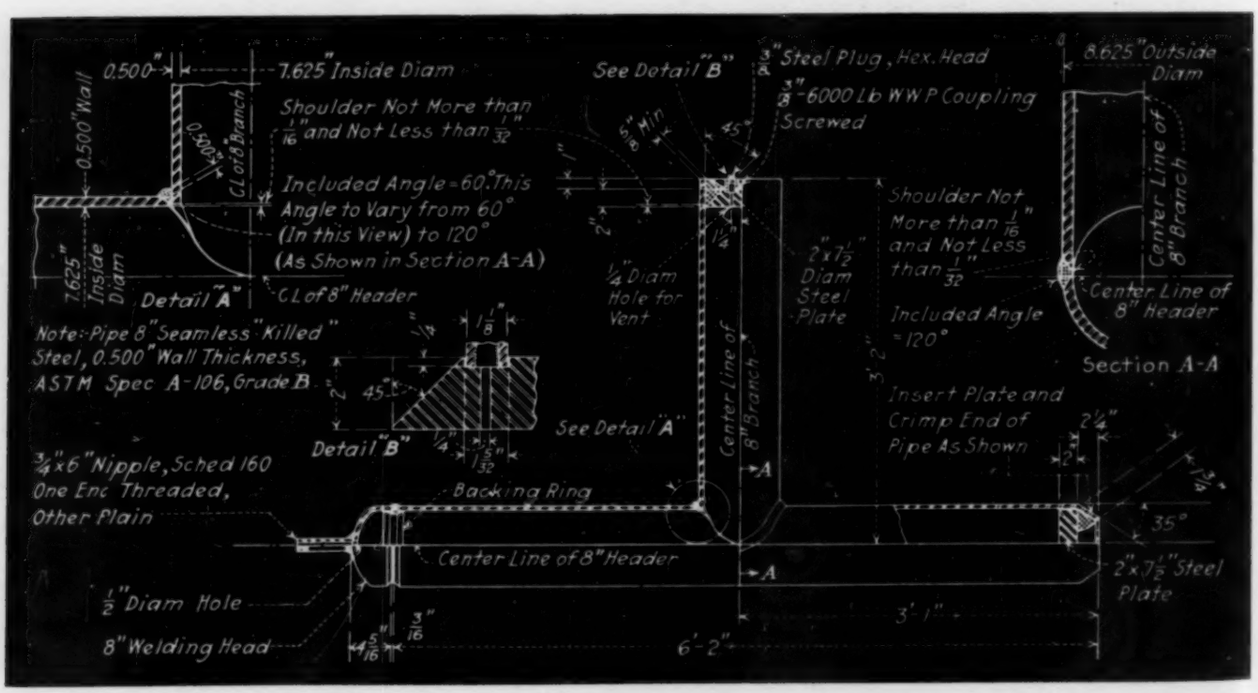
The testing procedure consisted principally in subjecting the manifold to hydrostatic pressure (the testing medium being automobile lubricating oil) and a series of about 1200 pressure cycles was imposed. Most of these ranged from 98 to 3510 lb per sq in., but about 50 tests were carried up to 5060 lb per sq in. Strains were determined at 50 stations in the vicinity of the welded juncture of branch and header by the use of tensometers, while overall distortion was measured by means of dial gages with 0.001 in. graduations. Following the series of tests on the unreinforced manifold (which included some tests employing lateral bending loads applied to the branch), the reinforcing ring was welded in place and the previous tests were repeated, using tensometer stations located radially outward from the original ones.

In testing the unreinforced manifold, stresses on the surface of the branch were highest near the flat portion on the side, where values of 50,000 to 60,000 lb per sq in. were found, these stresses representing the combined effect of bending of the wall and stretching by internal pressure. As regards end closures, it was determined that, under elastic conditions, their influence extends only a few inches from the ends.

According to the authors' summary, "the investiga-

This report was prepared for HEATING, PIPING AND AIR CONDITIONING by F. E. Wertheim, a member of the A.S.M.E.

Fig. 1—Details of test manifold


0.500" Wall
7.625" Inside Diam
Shoulder Not More than 1/16" and Not Less than 1/32"
Included Angle = 60°. This Angle to Vary from 60° (In this View) to 120° (As Shown in Section A-A)
C L of 8" Branch
C L of 8" Header
Detail "A"
Note: Pipe 8" Seamless "Killed" Steel, 0.500" Wall Thickness, ASTM Spec A-106, Grade B
Detail "B"
See Detail "B"
See Detail "A"
3/8" Steel Plug, Hex. Head
3/8" - 6000 Lb W W P Coupling Screwed
1/4" Diam Hole for Vent
2"x7 1/2" Diam Steel Plate
Shoulder Not More than 1/16" and Not Less than 1/32"
Included Angle = 120°
Insert Plate and Crimp End of Pipe As Shown
Backing Ring
Center Line of 8" Branch
Center Line of 8" Header
3/4"x6" Nipple, Sched 160 One End Threaded, Other Plain
1/2" Diam Hole
8" Welding Head
2"x7 1/2" Steel Plate
8.625" Outside Diam
Center Line of 8" Branch
Center Line of 8" Header
Section A-A
45°
35°
6'-2"
3'-1"

tion was partially successful in determining the magnitude and location of the stresses on the outer surface of the manifold, but was only of minor assistance in understanding the stress conditions on the inner surface. Even on the outer surface it was not possible to separate the effects of bending and of membrane stresses.

"The reinforcing ring was shown to assume its share of the load in the area which it covered but, as far as could be determined from strain gage measurements, had little significant effect on the stresses in the branch and header not covered by the ring. It even increased the measured strains in some places where the ring joined the header on the side of the manifold.

"While pronounced deformation or bulging was observed at the sides of the ring reinforced manifold at a pressure only 70 per cent of that which would be expected to cause yielding in a straight length of pipe, eventual failure occurred at a pressure approximately the same as would be expected to burst a straight length of pipe. It appears reasonable to conclude, therefore, that the ring type of reinforcement proportioned in accordance with present practice will develop the full strength of the pipe."

Tested to Destruction

After completion of the tests by Messrs. Everett and McCutchan, the manifold was turned over to the subcommittee on welded branch connections of A.S.A. Sectional Committee B31, who sent it to the research laboratories of the Crane Co. for the test to destruction. The results were reported in a discussion by E. R. Seabloom, research engineer, division of engineering and research. The first step was to stress relieve the manifold at 1200 F for removal of the effects of cold working due to the previous tests.

It was decided to supplement the strain gage readings by making dial gage measurements of deformations due to hydrostatic pressure, and the manifold was set up as shown in Fig. 3. Hydrostatic pressure was applied gradually in 400 lb increments and dial readings were recorded at each increment of pressure up to and including 4400 lb per sq in. Some of the deformations are expressed graphically in Fig. 4. It will be observed that

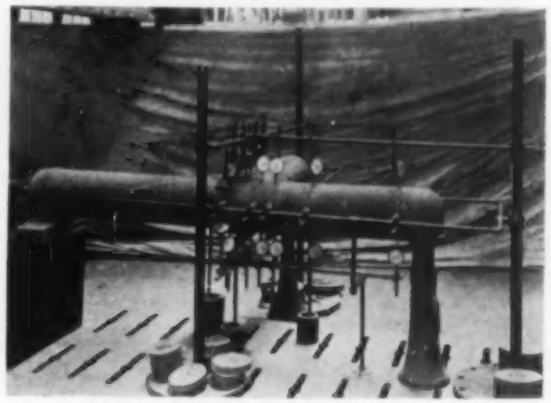


Fig. 3—Manifold set up for test

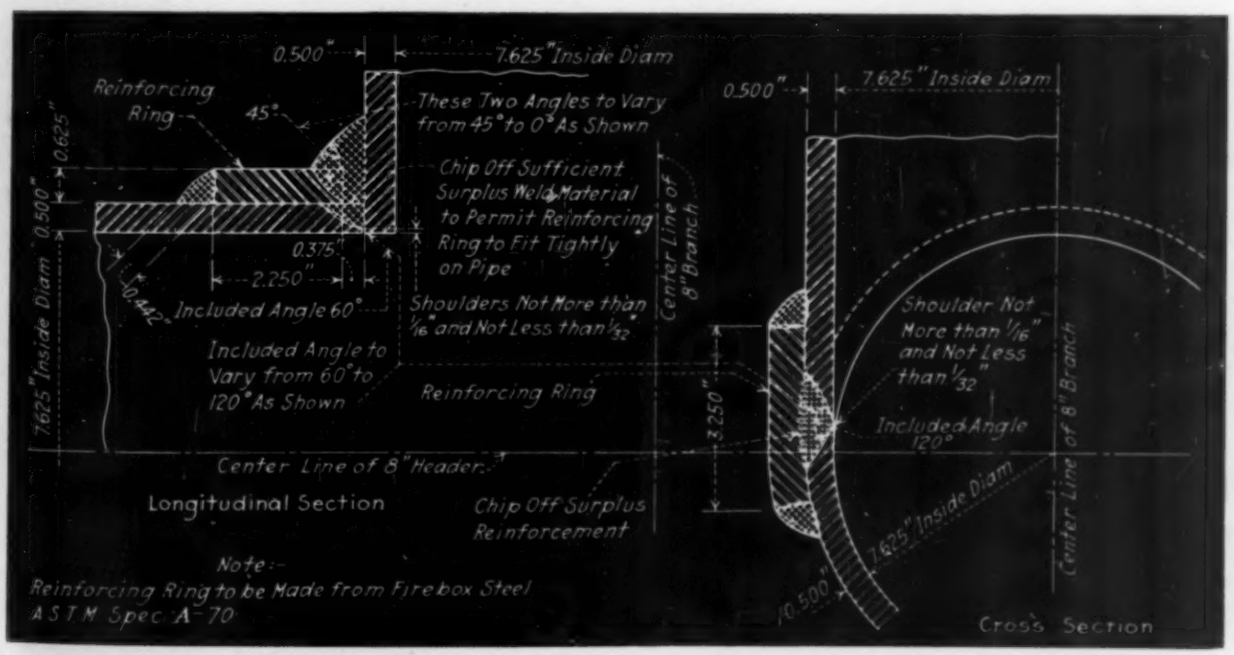
the maximum deflection occurred on the nozzle, immediately above the reinforcing ring at right angles to the run of the manifold, as indicated by dials 7 and 8. Considerable deformation also occurred adjacent to and in the reinforced area. This is due to the deviation from a circular cross section when a full-size nozzle is attached, producing a weaker structure to resist internal pressure.

After removal of the dials, the "test to destruction" was made. Internal pressure was increased gradually, failure finally occurring at 8750 lb per sq in. The burst manifold is shown with a series of similar casualties on page 117, where are also shown final tests on the end closures.

High Temperature Bolted Joints

Although welding is peculiarly adaptable in the high temperature high pressure field, separable joints remain

Fig. 2—Details of reinforcing ring


0.500"
7.625" Inside Diam
Reinforcing Ring
45°
These Two Angles to Vary from 45° to 0° As Shown
0.625"
0.500"
Chip Off Sufficient Surplus Weld Material to Permit Reinforcing Ring to Fit Tightly on Pipe
0.375"
2.250"
0.442"
Included Angle 60°
Shoulders Not More than 1/16" and Not Less than 1/32"
Center Line of 8" Branch
Included Angle to Vary from 60° to 120° As Shown
Reinforcing Ring
3.250"
7.625" Inside Diam
Center Line of 8" Header
Longitudinal Section
Chip Off Surplus Reinforcement
Note:-
Reinforcing Ring to be Made from Firebox Steel A.S.T.M. Spec. A-70
0.500"
7.625" Inside Diam
Shoulder Not More than 1/16" and Not Less than 1/32"
Included Angle 120°
Center Line of 8" Branch
7.625" Inside Diam
0.500"
Cross Section

a necessity in many apparatuses; for instance, valve bonnet and turbine or pump casing joints.

In his paper, "Analysis of Bolted Joints at High Temperatures," E. O. Waters, associate professor of mechanical engineering, Yale University, referred particularly to piping and the circumferential joints of pressure vessels. "Heretofore", said Professor Waters, "the design of bolted connections in this field has been based

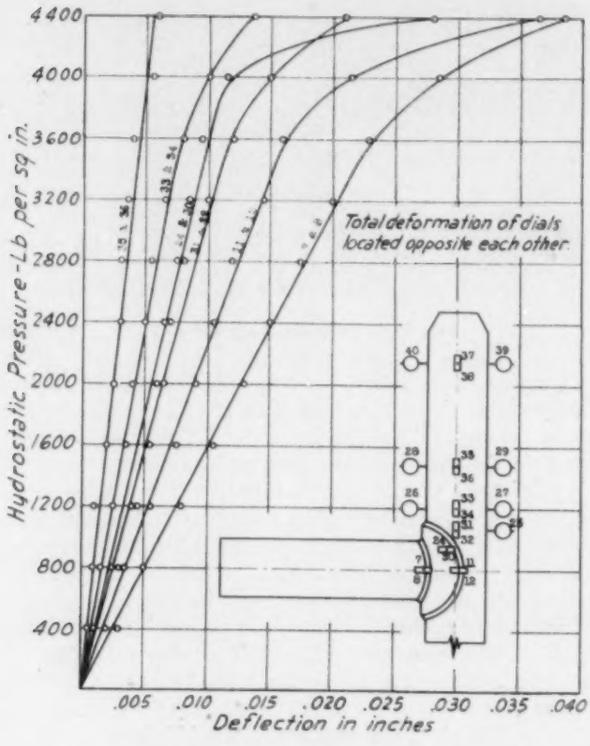


Fig. 4—Diametrical deflections of manifold and reinforcing ring

more or less approximately on the elastic theory. When, however, plastic flow of the material, due to service temperatures of 750 F or more, is superimposed upon the elastic displacements, the resultant stress distribution is at considerable variance with that prescribed by pure elasticity. Preliminary study shows that the flanges in a bolted pipe joint have a more nearly uniform, and hence more favorable, stress distribution after creep has occurred, than under purely elastic conditions. But since the tightness of the joint depends on the elastic stretch of bolts and flanges, a long service life at high temperatures would seem to preclude the use of increased design stresses." Professor Waters suggested that designers, instead of seeking to take advantage of the favorable stress distribution due to creep, had better develop or invent packing elements which do not depend upon the bolt tension and flange bending moment for tightness. However, although it was stated that the analysis given was preliminary, a complete mathematical analysis of the bolted joint at high temperatures has been developed.

Pipe Flanges Subjected to Creep

In a paper on "Stresses and Deformations in Pipe Flanges Subjected to Creep at High Temperatures," Joseph Marin, assistant professor of engineering materials, Rutgers University, gave an approximate analysis of the stresses and deformations produced in the flange part of temperature joints, *i.e.*, the stresses and deformations produced in a circular ring of rectangular cross section subjected to twisting couples uniformly distributed along its center line. The author considers this approximation to be justifiable in view of the complexity of the problem and the uncertainties still present in the fundamental creep-stress relation for metals. In his solution, a deflection theory is also developed for straight beams subjected to bending accompanied by creep.

Temperature Variations and Creep Strength

"The Effect of Temperature Variations on the Creep Strength of Steels" was analyzed by Ernest L. Robinson, turbine engineering department, General Electric Co., whose paper gave formulas and curves for the use of designers in solving problems which arise in predicting the effects of temperature change, variation, or fluctuation on the creep rate of steels at elevated temperatures.

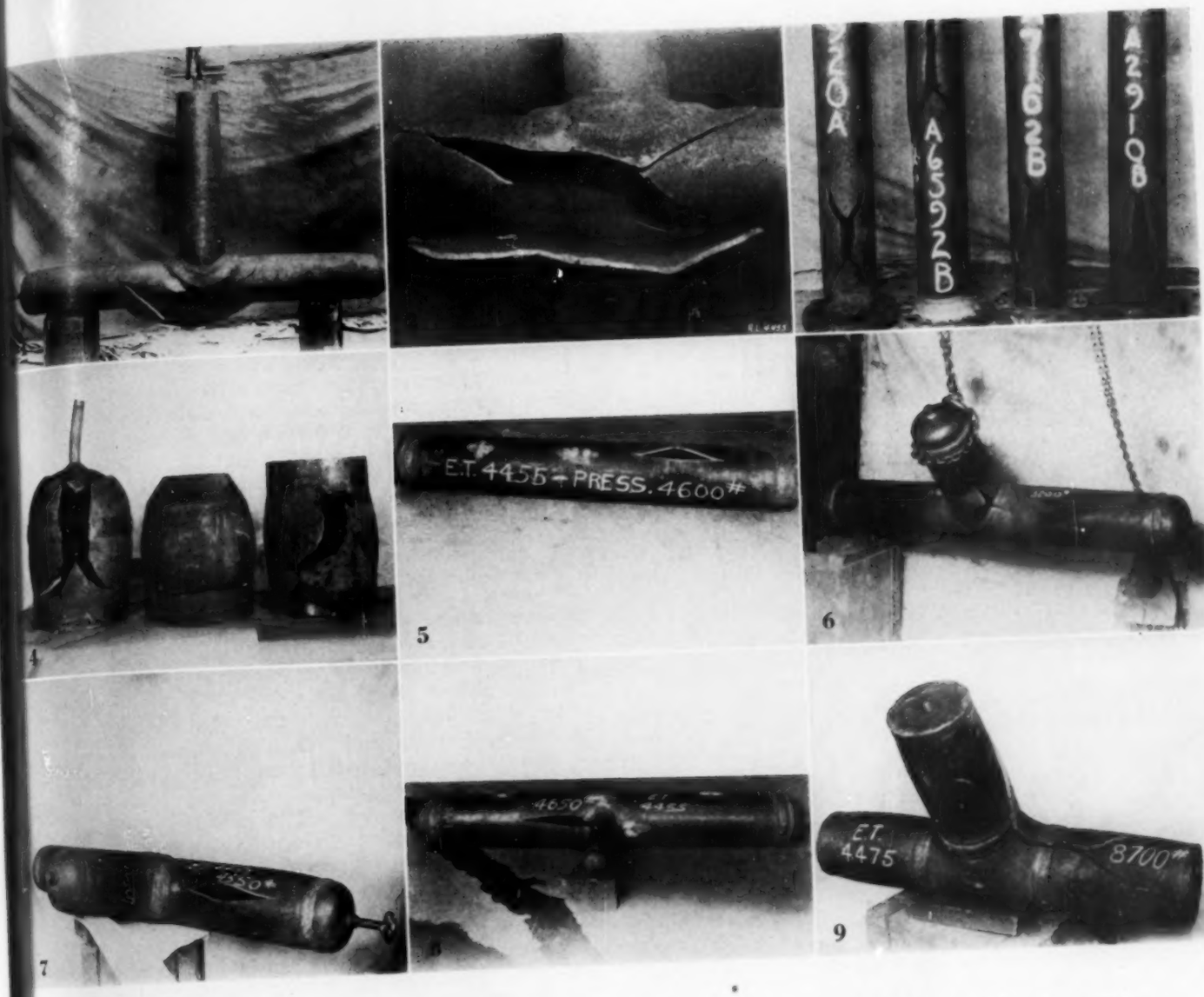
Referring to creep tests, Mr. Robinson emphasized the distinction that while a change in *constant* temperature level of 1 F may cause a change of creep rate of several per cent, it is very likely to require a fluctuation of as much as 10 or 20 F above, or with equal amount below, *average* temperature to produce as great a change in rate as that caused by the single degree alteration in the case of the constant temperature level.

The practical utility of the curves and formulas was illustrated by citing examples in which it was shown that when materials with certain creep properties were to be used in steam turbines or in superheaters where the temperature control allowed swings of 27 F above normal, an increase of creep rate would ensue which might halve the life of the equipment.

Strictly speaking, steam turbine design is not within the field of HEATING, PIPING AND AIR CONDITIONING; nonetheless, we believe the paper "Practical Aspects of Turbine Cylinder Joints" by C. B. Campbell, assistant division engineer, Westinghouse Electric & Manufacturing Co. is of particular value to our readers. Mr. Campbell has presented an extremely interesting article in which the principles underlying steam tight joint design are clearly stated. It is hoped to publish an abstract or the entire paper in a future issue.

Piping Post Mortems

On the next page is shown a series of pictures which illustrated the discussion of the Everett-McCutchan paper referred to above.



WE show herewith a series of pictures which were used to illustrate E. R. Seabloom's[1] discussion of the paper by F. L. Everett and Arthur McCutchan[2] on "Investigation of Stress Conditions in a Full Size Welded Branch Connection." Tests of this character afford a means of studying stress conditions and of locating maximum stress concentrations; they also furnish test data of value in high temperature, high pressure piping design. A brief summary of the data relating to these tests follows:

1—Test header used in Everett-McCutchan investigation. Failure at 8750 lb per sq in.

2—Enlarged detail of 1 indicates that the failure was possibly influenced by the holes drilled for tensometer.

3—Specimens cut from an 8 in. standard weight lap welded pipe 22 ft long. Typical of pipe failure due to bursting stresses. The bursting pressures varied from 4000 to 4400 lb per sq in.

4—Illustrates end closures which were cut from the header and branch shown in 1. Those at the ends ruptured at 10,500 lb pressure. That at the left is an ellipsoidal head, butt welded to the pipe. The right hand one is a flat head fused to the pipe wall for its entire thickness. The central closure consisted of a flat head inserted in the pipe which was crimped over and welded to the outer face of the head. Bursting pressure was 11,200 lb per sq in.; higher than the other specimens because a shorter span was subjected to pressure load.

5-8—Test specimens 5 ft long, all cut from the same length of 8 in. Schedule 40 A.S.T.M. Specification A106 Grade A carbon steel tubing. Rupture pressures (lb per sq in.) were as follows:

Straight pipe (5)	4600
Manifold with unreinforced 8 in. branch (6)	3200
Manifold with reinforced 8 in. branch (7)	4550
Manifold with unreinforced 4 in. nozzle (8)	4650

It is apparent that failure at the crotch is a characteristic of all the non-reinforced specimens, while the headers with full-size branch connections and ring reinforcement, as shown in 1 and 7, did not fail at the crotch, the reinforcement developing the full strength of the pipe.

9—Cast steel tee of 6 in. size welded to 6 in., Schedule 80, A.S.T.M. Specification A106 Grade A carbon steel pipe. Rupture pressure, 8700 lb per sq in.

[1]Research Engineer, Division of Engineering and Research, Crane Co.

[2]Presented at the annual meeting of the American Society of Mechanical Engineers, New York, N. Y., December 6 to 10, 1937.

E. Vernon Hill's* Personal Comments

Saturated Air Above 212 F

IN my "Comments" for October on "saturated air," I told about a graduate of a well known technical school who was discussing the subject of practical humidity in connection with warm air furnace installations, and whose remarks are of interest to readers of H. P. & A. C. because certain fundamentals of psychrometry are involved.

He made the statement that accurate control of the humidifier in the bonnet of the furnace was a prime essential, otherwise the moisture content of the air might rise to a point that would not only produce discomfort, but cause actual damage to the walls and furnishings of the building. He went on to say that air, at high temperatures, 300 or 400 F possibly, in a gravity system of heating the air, absorbs so much moisture that overhumidification must be guarded against.

I took issue with him after the discussion, and stated that high temperature air *carried* very little moisture. Dropping into unscientific vernacular I said, "In fact, air at 400 F could not possibly carry more than 1 per cent relative humidity." Some of the bystanders expressed their disbelief and the speaker laughed so long it almost made me wonder whether or not I had misread (or misinterpreted) the excellent tables and formulas in the "Guide." In my October "Comments" I said, "Perhaps some clear thinking, kind hearted 'psychromanic' will settle the controversy."

In response, I received the following letter from W. W. Howe:

"We can divide the proposition into a practical side and a theoretical side. Let us suppose that you are both talking about the ordinary warm air furnace, equipped with what is known as a humidity pan, usually located in the furnace bonnet, and that this pan is connected to city water supply so that a constant level is maintained in the pan by the float valve.

"If the temperature of the air in the bonnet reaches a temperature of 400 F, the pan is exposed to heat at this temperature or higher. Evaporation from this pan would be very rapid and steam would leave the pan at a temperature of 212 F and mix with the air stream. In the air stream, the steam would assume a superheat of 188 F and reach the same temperature as the air, or 400 F. This stream of air and superheated steam would reach the conditioned space via the warm air register and as soon as the superheated steam struck a cold surface at a temperature lower than the dew point, condensation would result. Certainly, if the quantity of water evaporated by the pan was excessive, the quantity condensed might easily be so great as to damage furnishings or decorations, just as the steam from a steam pipe would damage decorations or furnishings.

"It follows that the quantity of water being evaporated by the pan must be regulated, if it is desired to maintain a given relative humidity in the conditioned space. Your friend, the engineer, is certainly correct this far. It seems logical that the same thing would happen if the bonnet temperature was over the boiling point of the water, that is over 212 F, so that it is evident that the actual temperature of the air, that is any temperature over 212 F, had nothing to do with the excessive condensation. The air temperature could have been 213, or 313, or 413 F and the same damage would have occurred and in proportion to the evaporation of the pan and distinctly not in proportion to the air temperature.

"Now if we take the theoretical side and ask the question—How many grains of moisture would 1 lb of air at a temperature of 400 F and at atmospheric pressure hold, if saturated?—I must say that I do not know. If we look at a table of the properties of saturated air, we find a constantly decreasing quantity of air in the mixture of 1 cu ft of air and vapors to saturate until at a temperature of 212 F, the air is completely driven out and the cubic foot is all vapor.

"It seems logical, when thinking of the very great expansion of volume which takes place when water is converted into steam and the further expansion that takes place when the steam is superheated, that the actual quantity of vapor that could be present when mixed with high temperature air would be very, very small indeed. What the actual partial pressure of the superheated steam at 400 F and atmospheric pressure and what the saturation pressure under the same condition would be, I do not know and don't seem to find any data on the subject. If we had these figures, we could get the relative humidity. Lacking any real data, I am willing to back your statement that it would be less than 1 per cent.

"I am sure that you can give the necessary calculation for an exact answer and would be glad to have you do so."

In answer to Mr. Howe's inquiry for further information on this subject, I will say at this time, that relative humidity is a term that cannot be used in temperatures above 212 F, unless we are willing to adopt an entirely new definition for relative humidity. The subject however, is one of considerable interest and I hope to discuss it in some detail in March.

*Director of Research and Education, Refrigeration and Air Conditioning Institute. Member of Board of Consulting and Contributing Editors.

New Tables of Psychrometric Properties of Air-Vapor Mixtures

By William Goodman*

NEW psychrometric tables computed from the data in the new Keenan and Keyes steam tables were presented in the January issue. It is the purpose of the text not only to describe the use of the tables and how they were computed, but also to describe the method of computing the heat to be removed when cooling and dehumidifying air. In January, the enthalpy of air-vapor mixtures, enthalpy of water vapor, and the correct method of computing the heat to be removed were discussed.

Approximate Methods of Computing Heat to Be Removed

Approximate methods for computing the heat to be removed in cooling and dehumidifying air also use equations 5 and 5a. However, approximate values of the enthalpy h are used in these equations, instead of the true values computed by equations 6, 7, or 8.

There are only two commonly used methods of approximating the enthalpy, and both of them are based upon the fact that the enthalpy of an air-vapor mixture varies little for any *one* wet bulb temperature. That the true enthalpy of an air-vapor mixture is almost constant for any one wet bulb temperature will be evident from an inspection of the skeleton psychrometric chart, Fig. 2. In this chart the moisture content is plotted as the ordinate as in most charts, but the enthalpy is plotted along the abscissa. However, if instead of drawing the lines of constant enthalpy vertically, they are inclined at an angle as in Fig. 2, the resulting chart will have the familiar appearance of the customary psychrometric chart; that is, the lines of constant wet bulb temperature will slope downward from left to right, and the lines of constant dry bulb will be almost vertical.

In Fig. 2, the heavy line drawn through points 1, 2, and 3 represents any one line of constant wet bulb temperature. The three sloping parallel lines drawn through each of these points are lines of constant enthalpy. The distances between these three parallel enthalpy lines, and the angle between them and the line of constant wet bulb temperature, is exaggerated somewhat for the sake of clarity. Actually, the distances and the angle would be so small that the wet bulb line would almost coincide with all three enthalpy lines. Bearing this in mind, it is evident from Fig. 2 that, although the enthalpy will be different for every point on the constant wet bulb line, the difference will be small. Therefore, all points lying on any one wet bulb line may be said to have approximately the same enthalpy. (For further discussion of this point, see appendix 4).

If the true enthalpy is known at any one point on a line of constant wet bulb temperature, it can be used as the approximate enthalpy of all other points on that same wet bulb line. Now, of all the possible points along any wet bulb line, the enthalpy values at only two points have been tabulated. These are point 1 for saturated air, and point 3 for dry air (zero moisture content). The

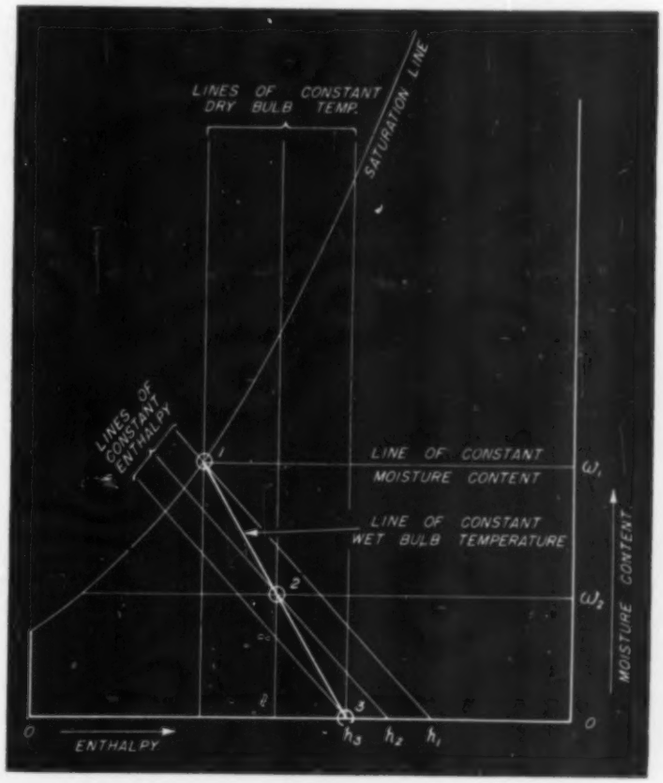


Fig. 2—Skeleton psychrometric chart showing that true enthalpy is almost constant for any one wet bulb temperature. (Angle between the wet bulb line and enthalpy lines, and distances between the enthalpy lines, are exaggerated for the sake of clarity)

values of the enthalpy of saturated air are tabulated in column 11 of the tables of properties of mixtures of air and saturated water vapor published in January, for various wet bulb temperatures. The enthalpy at point 3 of any line of wet bulb temperature is shown in appendix 5 to be equal to the sigma function, which was mentioned on page 3 of the January article. Values of the sigma function have been tabulated for various wet bulb temperatures by Carrier and Mackey.[2]

Tabulated values of the enthalpy at either point are equally convenient to use in approximate computations. This being the case, the choice between them should be

*The Trane Co. Member of Board of Consulting and Contributing Editors.

Part 2. Part 1 was published in HEATING, PIPING AND AIR CONDITIONING, January, 1938, pp. 1-7.

Copyright, 1938, by William Goodman.

[2]See H. P. & A. C., January, 1938, p. 3.

based on the accuracy of the results obtained by their use. Therefore, examples 1 and 2, previously given, were also computed by the two approximate methods, and the results are summarized in the table herewith.

Summary of Results Obtained by Solving Examples 1 and 2 by Correct and Approximate Methods

(For details of computations, see Appendix 6)

Values of Enthalpy Used	Equation Used	Example 1 Condensate Remains *Liquid*		Example 2 Condensate *Freezes*	
		q	Deviation	q	Deviation
True Enthalpy	5	14.976	0%	32.389	0%
	5a	15.044	+0.45%	30.882	−4.4%
Enthalpy of saturated air at wet bulb temp.	5	15.272	+1.98%	32.687	+1.2%
	5a	15.34	+2.43%	31.28	−3.1%
Σ Function	5	14.672	−2.03%	31.537	−2.3%
	5a	14.74	−1.58%	30.13	−6.7%

Although this table gives the results of only two examples chosen at random, it is nevertheless indicative of the results that may be expected generally. The percentage deviation with either method of approximation is usually about the same, but a negative deviation is always obtained when the sigma function is used for computing problems in cooling and dehumidification. It is evident that there is little to choose between these two approximate methods as regards accuracy. This being the case, the enthalpy of saturated air is preferable for approximate computations because its use does not require any new definitions; enthalpy is a function that is used in all other fields of thermodynamics.

Computation of Absolute Humidity

The absolute humidity, tabulated in column 4 of the tables, is the weight of moisture mixed with each pound of dry air. Above 32 F, it was computed by means of the following equation derived from Dalton's law of partial pressures:

$$w = \frac{53.34}{144}\,\frac{Td}{(P-p_v)} \qquad [9]$$

The partial vapor pressure, p_v, and the density of the vapor d were obtained from the Keenan and Keyes steam tables.

Below 32 F, the absolute humidity was computed from the well-known equation:

$$w = 0.622\,\frac{p_v}{P-p_v} \qquad [10]$$

Equation 10 is derived from [9] on the assumption that the water vapor at low pressures behaves as a perfect gas. However, [10] can be used with little error for computing the absolute humidity at temperatures above 32 F. For instance, at the highest temperature covered by the psychrometric tables, 200 degrees, the absolute humidity computed by [10] is 2.262 lb, whereas by [9] it is 2.292 lb, a deviation of 1.3 per cent. At 100 F, the deviation is considerably less, being only about ¼ of 1 per cent.

Below 32 F, the vapor pressure of subcooled water was used in equation 10. These vapor pressures were computed from the equations derived by Washburn[3] for the International Critical Tables. These equations were also used by Keenan and Keyes for the computation of an abbreviated table of the properties of water vapor below 32 F (published in their steam tables), although they state, "The primary data are not as consistent as would be desirable for many purposes. . . ."

Vapor pressures over subcooled water were used in equation 10 for computing the absolute humidities below 32 F. Theoretically, vapor pressures over ice should have been used in computing the absolute humidity because air at subfreezing temperatures is usually brought in contact with an ice-covered surface during some part of the conditioning cycle. However, Keyes and Smith[4] state, "the water content of air in equilibrium with ice does not correspond to what would be expected on the basis of Dalton's law from the known vapor pressures of ice, but rather to the pressures for subcooled water." The use of the vapor pressures of subcooled water, instead of those of ice, appears to be an empirical method of correcting for the defect of Dalton's law at temperatures below 32 F. The absolute humidities computed in this way are still to be regarded as those of air-vapor mixtures in contact with ice.

The values of absolute humidity tabulated in column 4 of the tables are larger than those which would be obtained if the vapor pressures over ice were used. Although the actual difference in the absolute humidity computed by means of either of the vapor pressures is always small, the percentage deviation becomes quite large at the lower temperatures. Thus at zero degrees, the deviation in the absolute humidity computed by the two methods is 18.9 per cent, while at 40 F below zero it is 47.0 per cent. In the light of these large deviations, there would appear to be need for a thorough experimental investigation of the absolute humidity of air at low temperatures. The best that can be said for the tabulated data, which were computed by using the vapor pressures of subcooled water, is that they appear, in the light of the Keyes and Smith statement, to be more in accord with the facts than if the vapor pressures over ice were used.

Specific Volume of Dry Air and of Air-Vapor Mixtures

The specific volumes of *dry* air, tabulated in column 6, were computed by the usual perfect gas formula.

$$v_D = \frac{53.34}{144}\,\frac{T}{P} \qquad [11]$$

Instead of tabulating the volumes of *saturated* air in the psychrometric tables, factors are tabulated in column 7 which permit the volume of air having any degree of saturation to be computed quickly.

The derivation of the equation by which these factors were computed is given in appendix 7. These factors depend only upon the dew point temperature of

[3] "The Vapor Pressure of Ice and of Water Below the Freezing Point," by E. W. Washburn, *Monthly Weather Review*, October, 1934.

[4] "The Present State of Psychrometric Data," by Keyes and Smith. *Refrigerating Engineering*, March, 1934.

Symbols for Psychometric Tables and Text

d	Density of vapor	Lb per cu ft
h	Enthalpy of air-vapor mixture	Btu per lb of dry air
h'	Enthalpy of a *saturated* mixture at the wet bulb temperature	Btu per lb of dry air
h''	Enthalpy of a *saturated* mixture at the dew point temperature	Btu per lb of dry air
h_a	Enthalpy of air only, in an air-vapor mixture	Btu per lb of dry air
h_f	Enthalpy of water or ice at the dry bulb temperature	Btu per lb of steam
h'_f	Enthalpy of water or ice at the wet bulb temperature	Btu per lb of water
h''_f	Enthalpy of water or ice at the dew point temperature	Btu per lb of water
h_g	Enthalpy of vapor at the dry bulb temperature	Btu per lb of steam
h'_g	Enthalpy of vapor at the wet bulb temperature	Btu per lb of steam
h''_g	Enthalpy of vapor at the dew point temperature	Btu per lb of steam
h_0	Enthalpy of make-up water supplied to, or of condensate or ice removed from, air conditioning apparatus	Btu per lb of water
h_v	Enthalpy of vapor only, in an air-vapor mixture	Btu per lb of dry air
p_v	Vapor pressure of water	Lb per sq in.
P	Total barometric pressure	Lb per sq in.
q	Heat removed from air-vapor mixture by external means, such as chilled water or direct expansion refrigerant	Btu per lb of dry air
r'	Enthalpy of either vaporization or sublimation at the wet bulb temperature	Btu per lb of steam
s	Humid specific heat of air-vapor mixture	Btu per lb of dry air per deg Fahrenheit
s'	Humid specific heat of a *saturated* mixture at the wet bulb temperature	Btu per lb of dry air per deg Fahrenheit
s''	Humid specific heat of a *saturated* mixture at the dew point temperature	Btu per lb of dry air per deg Fahrenheit
t	Dry bulb temperature of air-vapor mixture	Deg Fahrenheit
t'	Wet bulb temperature of air-vapor mixture	Deg Fahrenheit
t''	Dew point temperature of air-vapor mixture	Deg Fahrenheit
T	Absolute temperature	Deg Fahrenheit
v	Specific volume of air-vapor mixture	Cu ft per lb of dry air
v_D	Specific volume of dry air	Cu ft per lb of dry air
w	Absolute humidity (weight of moisture in an air-vapor mixture. Also called the moisture content)	Lb per lb of dry air
w'	Absolute humidity at the wet bulb temperature	Lb per lb of dry air

Numerical Subscripts—Numerical subscripts (such as 1 and 2) are used to indicate the point in the air conditioning cycle at which the quantity is to be taken. For example, t_1 indicates the initial dry bulb temperature of the air; t'_1, the initial wet bulb; t_2 the final dry bulb; t'_2, the final wet bulb; h_1, the initial enthalpy of the air; w_2 the final absolute humidity; and h'_2, the enthalpy of saturated air at the final wet bulb temperature.

the air. When they are multiplied by the specific volume of dry air at the dry bulb temperature, the increase in the specific volume due to the vapor is obtained. The use of these factors is illustrated by example 3.

Example 3: Find the specific volume of an air-vapor mixture having a dry bulb temperature of 70 F if (*a*) the mixture is saturated, and (*b*) the dew point temperature is 51 F.

Solution:

(*a*) From column 6, for a dry bulb temperature of 70 F, $v_D = 13.35$ cu ft per lb.

From column 7, for a dew point temperature of 70 F, volume increase = 2.53 per cent.

Therefore, specific volume of saturated air at 70 F is,

$$v = 13.35 + 13.35 \frac{2.53}{100}$$
$$= 13.35 + 0.34$$
$$= 13.69 \text{ cu ft per lb}$$

(*b*) For a dew point temperature of 51 F, volume increase = 1.27 per cent.

Therefore, specific volume of air having a dry bulb temperature of 70 F and a dew point temperature of 51 F is

$$v = 13.35 + 13.35 \frac{1.27}{100}$$
$$= 13.35 + 0.17$$
$$= 13.52 \text{ cu ft per lb}$$

The editors are privileged to present new and accurate tables of the psychrometric properties of air-vapor mixtures, which have been compiled by Mr. Goodman, based upon the new Keenan-Keyes steam tables. . . . In his discussion of their use, the author clarifies the subject of heat balances, which is basic to all air conditioning computations

It is interesting to note from column 7 that the increase in volume for humid air is less than 1 per cent for all dew point temperatures below 45 degrees.

Humid Specific Heat

The humid specific heat, values of which are tabulated in column 8, is defined and computed by

$$s'' = 0.24 + 0.45w \quad \ldots\ldots [12]$$

In addition to the uses which have previously been made of this quantity in computing enthalpy, it is also useful in the Carrier[5] equation for computing the wet bulb temperature.

$$s''(t - t') = r'(w' - w) \quad \ldots\ldots [13]$$

For evaporation from an ice coated surface (such as a wet bulb thermometer with a frozen wick), the enthalpy of sublimation—not of vaporization—should be used for r' in [13]. Values of the enthalpy of sublimation are given in the small table of "Enthalpy of Ice and Low Temperature Water Vapor," page 2, January issue.

On the other hand, if the water on the wet bulb wick is subcooled—not frozen—the enthalpy of vaporization should be used, r' in [13]. Values of the enthalpy of vaporization for subcooled water may be found accurately by means of the following empirical formula:

$$r = 1093 - 0.55t \quad \ldots\ldots [14]$$

This formula is obtained by subtracting the enthalpy of the liquid, $t - 32$, from equation 4.

[Concluded]

Appendix 1—Derivation of Equation 6

Equation 3 of the text is:

$$h = 0.24t + wh_g$$
$$= 0.24t'' + 0.24(t - t'') + wh_g \quad \ldots\ldots [a]$$

From a consideration of [4] of the text, it is evident that

$$h_g - h_g'' = 0.45(t - t'')$$
$$h_g = h_g'' + 0.45(t - t'') \quad \ldots\ldots [b]$$

Substituting into [a] and reassembling terms,

$$h = (0.24t'' + wh_g'') + (0.24 + 0.45w)(t - t'') \quad \ldots\ldots [c]$$

Now if the air is saturated at the dew point temperature,

$$h'' = 0.24t'' + wh_g''$$

Also, by equation 12 of the text,

$$s'' = 0.24 + 0.45w$$

Making these substitutions, [c] reduces to equation 6 of the text

Appendix 2—Derivation of Equation 7

For simplicity the air conditioning apparatus of Fig. 1 is assumed to be an air washer in which the water is continuously recirculated and no heat is added or subtracted by external means. In other words, the process is adiabatic. In such a case, $q = 0$.

As water evaporates into the air, it must be replaced in a continuous process. For make-up water, the direction of the arrow marked "condensate" in Fig. 1 must be reversed so that it points into the washer.

In an adiabatic process in a perfect washer, the air will leave the apparatus saturated at a temperature equal to the initial wet bulb temperature of the air-vapor mixture. Furthermore, to satisfy the requirements of an adiabatic process, the make-up water must be supplied at the temperature at which it will evaporate in the washer. This temperature is also equal to the initial temperature of adiabatic saturation. Therefore, in Fig. 1,

$$h_2 = h_1' \quad \ldots\ldots [a]$$
$$w_2 = w_1' \quad \ldots\ldots [b]$$
$$h_0 = h_{t1}' \quad \ldots\ldots [c]$$

[5] "Rational Psychrometric Formulae," by W. H. Carrier, *Transactions*, A.S.M.E., Vol. 33, 1911, p. 1005.

Remembering that the condensate arrow is reversed for make-up water, and that $q = 0$, the heat balance equation for Fig. 1 may be written as

$$h_2 = h_1 + (w_2 - w_1)h_0$$

Substituting [a], [b] and [c],

$$h_t' = h_1 + (w_1' - w_1)h_{t1}'$$

Dropping the subscript 1, and writing this equation in its general form, results in [7] of the text.

Appendix 3—Derivation of Equation 8

Equation 8 of the text is obtained by eliminating w between equations 3 and 7 of the text.

From [3] of the text,

$$w = \frac{1}{h_g}(h - 0.24t) \quad \text{.....[a]}$$

From [7] of the text,

$$w = w' - \frac{1}{h_t'}(h' - h) \quad \text{.....[b]}$$

Equating [a] and [b], and reassembling terms,

$$h = 0.24t + w'h_g - \frac{h_g}{h_t'}(h' - h) \quad \text{.....[c]}$$

Now,

$$0.24t = 0.24(t - t') + 0.24t' \quad \text{.....[d]}$$

From a consideration of [4] of the text, it is evident that

$$w'h_g = w'[h_g' + 0.45(t - t')] = w'h_g' + 0.45w'(t - t') \quad \text{.....[e]}$$

Substituting [d] and [e] into [c] and reassembling terms,

$$h = (0.24 + 0.45w')(t - t') + (0.24t' + w'h_g') - \frac{h_g}{h_t'}(h' - h)$$

But,

$$s' = 0.24 + 0.45w'$$

and $$h' = 0.24t' + w'h_g'$$

Substituting and rearranging terms,

$$h\left(1 - \frac{h_g}{h_t'}\right) = h'\left(1 - \frac{h_g}{h_t'}\right) + s'(t - t')$$

Solving this equation for h, equation 8 of the text is obtained.

Appendix 4—Proof That Enthalpy Is Almost Constant Along a Line of Constant Wet Bulb Temperature

Equation 8 can be used to show that the enthalpy of an air-vapor mixture changes little as long as the wet bulb temperature remains constant.

In equation 8, the numerical value of $\left[\frac{s'}{\frac{h_g}{h_t'} - 1}\right]$, which is the coefficient of the second term, is small. Hence, the numerical value of the second term itself will also be small as long as the difference between the dry and wet bulb temperatures of the air is moderate. This is usually the case for ordinary work. Therefore, by [8], it is apparent that the numerical value of the enthalpy of an air-vapor mixture will usually deviate but little from that of the enthalpy of saturated air at the same wet bulb temperature.

Appendix 5—Proof that $\Sigma_3 = h_3$ (Fig. 2)

For any given wet bulb temperature, the enthalpy of dry air is equal to sigma function. In other words, the enthalpy of air in the state 3 of Fig. 2 is equal to the sigma function for the wet bulb temperature through that point. In order to prove this statement, it will be shown that equation 5, when applied to a point on the one wet bulb line through point 3 of Fig. 2, can be reduced to Carrier's definition of the sigma function.

Writing equation 5 for the points 2 and 3 which lie on the same line of wet bulb temperature,

$$q = (h_2 - h_3) - (w_2 - w_3)h_0$$

But for an adiabatic process (that is, one along a line of constant wet bulb temperature as in Fig. 2),

$$q = 0, \text{ and } h_0 = h_{t2}' = h_{t3}'$$

Also at point 3, $w_3 = 0$

Therefore,

$$h_3 = h_2 - w_2h_{t2}' \quad \text{.....[a]}$$

Substituting equation 3 of the text for h_2,

$$\begin{aligned} h_3 &= 0.24t_2 + w_2(h_{g2} - h_{t2}') \\ &= 0.24t_2 + w_2[h_{g2}' - h_{t2}' + 0.45(t_2 - t_2')] \\ &= 0.24t_2 + w_2[r_2' + 0.45(t_2 - t_2')] \\ &= 0.24t_2 + w_2r_2' + 0.45w_2(t_2 - t_2') \quad \text{.....[b]} \end{aligned}$$

But according to the definition of Carrier,

$$\Sigma_3 = \Sigma_2 = 0.24t_2 + w_2r_2' + 0.45w_2(t_2 - t_2') \quad \text{.....[c]}$$

Therefore, as [b] and [c] are identical,

$$\Sigma_3 = h_3 \quad \text{.....[d]}$$

Also, substituting [d] into [a],

$$\Sigma_3 = \Sigma_2 = h_2 - w_2h_{t2}'$$

Dropping subscripts to make the equation more general,

$$\Sigma = h - wh_t' \quad \text{.....[e]}$$

This is the definition of the sigma function in terms of the enthalpy of an unsaturated air-vapor mixture. If the air is saturated, as at the point 1 of Fig. 2, [e] reduces to,

$$\Sigma = h' - w'h_t' \quad \text{.....[f]}$$

Appendix 6—Details of Computations for Examples 1 and 2 by Approximate Methods

	Example 1		Example 2	
	By Enthalpy of Sat. Air	By Σ Function	By Enthalpy of Sat. Air	By Σ Function
h'_1	38.53		38.53	
h'_2	23.19		7.25	
w'_1		0.01876		0.01876
h'_{t1}		43		43
$\Sigma_1 = h'_1 - w'_1h'_{t1}$		37.72		37.72
w'_2		0.0092		0.00229
h'_{t2}		23.0		−149.05
$\Sigma_2 = h'_2 - w'_2h'_{t2}$		22.98		7.59
$h'_1 - h'_2$	15.34		31.28	
$\Sigma_1 - \Sigma_2$		14.74		30.13
$(w_1 - w_2)\,h_0$	0.06826	0.06826	−1.407	−1.407
q by Eq. 5	15.272	14.672	32.687	31.537
q by Eq. 5a	15.34	14.74	31.28	30.13

Appendix 7—Increase in Volume of Dry Air Because of Moisture

The percentage increase in the volume of dry air due to the addition of moisture was computed by means of equation [d] given below.

By the perfect gas laws,

$$v_0 = \frac{53.34}{144}\frac{T}{P} \quad \text{.....[a]}$$

$$v = \frac{53.34}{144}\frac{T}{P - p_v} \quad \text{.....[b]}$$

Subtracting [b] from [a], and dividing by [b], the equation for the increase in the volume of dry air when moisture is added is obtained.

$$\text{Increase in volume of dry air} = \frac{p_v}{P - p_v} \quad \text{.....[c]}$$

or

$$\text{Percentage increase in volume of dry air} = 100\frac{p_v}{P - p_v} \quad \text{.....[d]}$$

It is evident that the increase in the air volume is a function of only the vapor pressure, which in turn depends only upon the dew point temperature of the air-vapor mixture.

Bibliography of Engineering Subjects

The Engineers' Council for Professional Development, 29 W. 39th St., New York, N. Y., has published a "Selected Bibliography of Engineering Subjects" prepared by the committee on professional training. There are five sections, as follows: Section I, Mathematics, Physics, Mechanics; Section II, Aeronautical Engineering, Civil Engineering; Section III, Chemistry, Chemical Engineering, Industrial Engineering; Section IV, Electrical Engineering, Mechanical Engineering; Section V, Metallurgy, Mineralogy and Geology, Mining Engineering.

In their preparation, more than one hundred teachers of technical subjects and professional engineers were consulted, and the final selection is therefore a compromise between the several recommendations. The bibliography represents books of college grade mostly, with annotations to indicate the nature and depth of the subject matter.

The pamphlets are available from the secretary of the Council for 10c per section (or 5c in quantities of fifty or more).

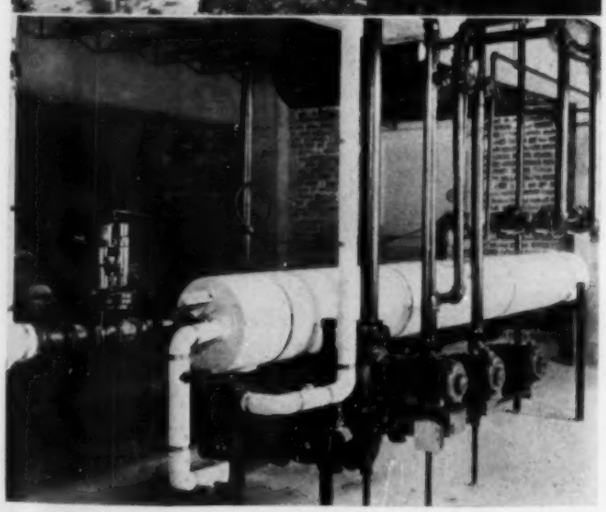
Fourteen Zone Hot Water System Heats Tuberculosis Sanatorium

By J. C. Pastor*

FLORIDA'S new 450 bed tuberculosis sanatorium, situated on a high plot of ground about nine miles west of Orlando, is heated with a 14 zone, forced circulation, single main, hot water heating system with some 15,000 sq ft of concealed copper radiation. The two major considerations that influenced the selection of this type of system were economy in first cost and operation, and the elimination of practically all piping from the basement, which makes more space available for other purposes. The building is of fireproof construction throughout with a light buff stucco exterior finish. Philip F. Kennard, St. Petersburg, was the designing architect and supervised the construction as well.

Fig. 1 shows the shape of the building in outline. There is a full basement under zones B, C, and D, the remainder being unexcavated, but the basement is unheated. Provision is made, however, for future heating by simply attaching a system of piping and unit heaters or radiators to the present converters.

***Above*—Front elevation of 450 bed tuberculosis sanatorium heated by 14 zone hot water system . . . *Below*—One of the five converters supplied with 50 lb steam and furnishing hot water to the concealed radiation**



The building is divided into 14 heating zones, as indicated on Fig. 1, with a total of approximately 15,000 sq ft of concealed convectors with removable fronts and top and bottom grilles.

By dividing the building into 14 zones, it was possible to keep pipe sizes small, five zones having 2½ in. mains and nine having 2 in. mains. There are five converters, located in the basement where shown, consisting of heaters with 12 in. diameter steel shells. Steam at 90 lb pressure is generated in a separate plant to supply steam at 50 lb pressure to sterilizers, steam tables, kitchen equipment, warming devices, etc. For this reason it was decided to use this 50 lb steam for the converters and thus have only one system of steam piping.

Steam is admitted to the converters through regulator valves which have adjustable ranges of 160 to 220 F. The bulb controlling the regulator is located in the supply header as it leaves the converter and therefore responds very quickly to temperature change in the circulating water when the pump starts. Condensate from the converter is returned to the boiler room through a bucket trap and a system of return piping. Each converter is provided with an expansion tank and relief valve, with the tank so arranged that practically all air is vented into it.

Each converter supplies hot water for three zones with a separate supply and return main and a booster pump for each zone. Flow control valves are used in both the supply and return mains to prevent gravity circulation when the pump is not running.

The Piping Layout

Fig. 2 shows a floor plan of zone 1B which is typical of the others. The supply main rises from the converter to the bar joist space (see Fig. 3) where it splits and parallels the outside walls to serve the radiators in the various rooms. The returns were brought below the ceilings in utility rooms and equipped with thermometers and balancing valves before being tied together and then dropped, in a chase in the partition, to the converter in the basement. The piping is supported on hangers with concrete inserts in the floor slab and does not rest on the bar joists. Where beams were encountered that were acting only as struts, the pipe was carried through the beam in a cast iron sleeve. Care was taken in making the layout to avoid load bearing beams, and in the two cases where they were encountered, the piping was dropped below the beam and then back to the former level with proper precaution to vent at these points.

Fig. 3 shows a typical radiator recess and the method of running the piping in the bar joist space, close to the

*Bell & Gossett Co.

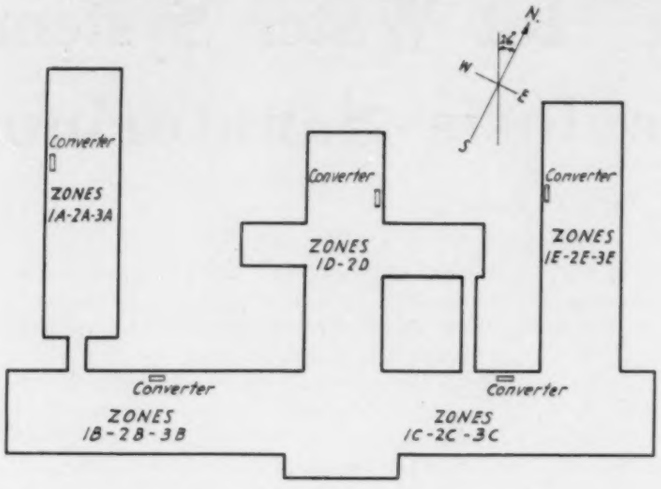


Fig. 1

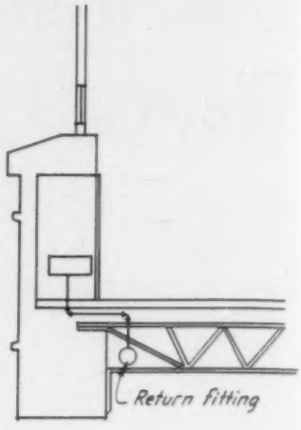


Fig. 3

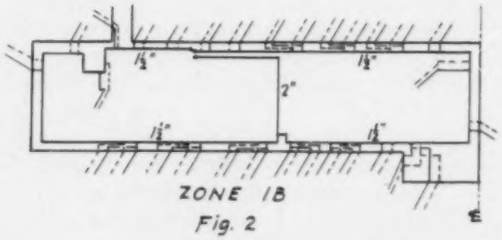


Fig. 2

Fig. 1—Plan showing arrangement of zones. . . . Fig. 2—Plan of a typical zone. . . . Fig. 3—Typical radiator recess, showing method of running piping in joist space

outside walls. All convectors are equipped with dampers and the air valves were brought out flush with the lower grille to facilitate air elimination.

Each booster pump is controlled by a line voltage thermostat with adjustment lock, located in the corridor at the nurses' station in each zone. The system was designed for a Btu emission of 240 at the radiators with a 20 F temperature drop. This, of course, can be varied at will by changing the temperature setting of the regulators.

[Concluded from page 107]

nished on the compressor by the manufacturer. This illustrates very well the futility of guessing at pipe sizes.

Now, let us assume that instead of a simple system with only one evaporator, that the system in question comprises three evaporators—one of 25 tons, one of 50 tons and one of 120 tons, operating at 290.8 lb gage pressure and 0 F saturated. How should the suction main be graduated in size?

For the first evaporator, 25 × 64.8 = 1620 fpm. This is satisfactory for a 2 in. suction where the limit is 1650 fpm.

For the second evaporator, 50 × 28.9 = 1445 fpm. This is satisfactory for a 3 in. branch suction where limit is 1750 fpm.

Joining these two, we have 75 tons capacity. 75 × 21.5 = 1612 fpm. This is satisfactory for a 3½ in. main suction where the limit is 1800 fpm.

For the third evaporator, 120 × 16.6 = 1992 fpm. This is too high for 4 in. where the limit is 1850 fpm. 120 × 10.5 = 1260 fpm. This is satisfactory for a 5 in. branch suction where the limit is 1900 fpm.

Joining this to the main, we have 195 tons capacity. 195 × 10.5 = 2050 fpm, which is too high for 5 in. where the limit is 1900 fpm. 195 × 7.42 = 1448 fpm, which is satisfactory for 6 in. where the limit is 1950 fpm.

Therefore the suction line should be graduated in size 6, 3½ and 2 in. with a

2 in. branch suction line for the first evaporator of 25 ton capacity;

3 in. branch suction line for the second evaporator of 50 ton capacity;

5 in. branch suction line for the third evaporator of 120 ton capacity;

regardless of the size suction connection on the compressor. If either the main suction or branch suction lines are extremely long, it is best to use the next larger nominal size pipe rather than that calculated from the table, especially if the actual velocity of vapor is very close to the design limit, in order not to cause too high a pressure drop.

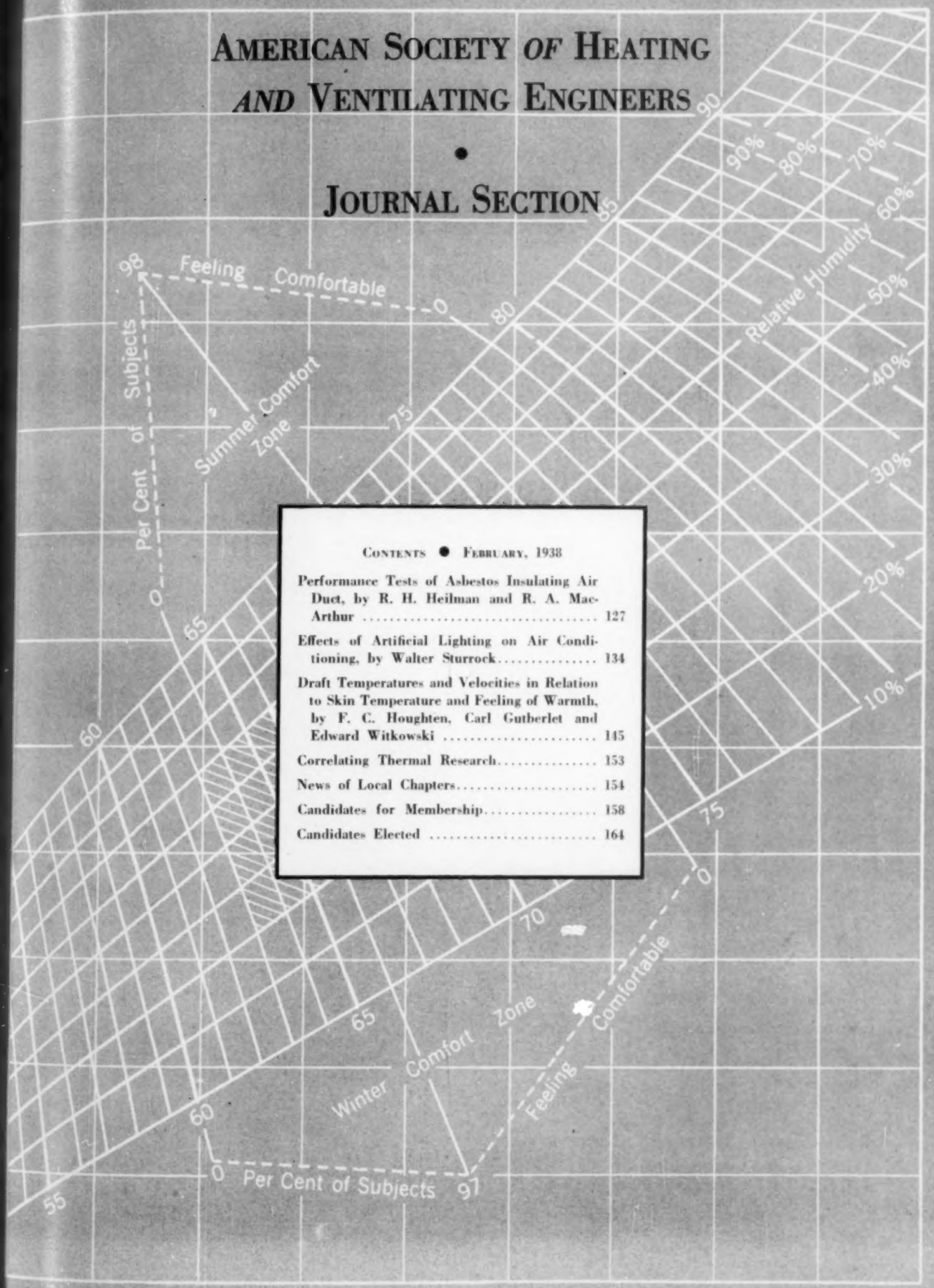
Even with these relatively low gas velocities, the vapor is so extremely heavy that sharp elbows, tees, offsets, etc. must be avoided. Because of the weight of the vapor, the momentum of the fluid stream is a considerable item and will frequently set up a bad pipe vibration due to a slight change in direction of flow. This frequently requires three- or four-way bracing at bends in the main suction line rigidly welded to the pipe as an anchor.

Heating · Piping and Air Conditioning

American Society of Heating and Ventilating Engineers

•

Journal Section



Contents • February, 1938

Performance Tests of Asbestos Insulating Air Duct, by R. H. Heilman and R. A. MacArthur 127

Effects of Artificial Lighting on Air Conditioning, by Walter Sturrock..... 134

Draft Temperatures and Velocities in Relation to Skin Temperature and Feeling of Warmth, by F. C. Houghten, Carl Gutberlet and Edward Witkowski 145

Correlating Thermal Research..... 153

News of Local Chapters..... 154

Candidates for Membership..... 158

Candidates Elected 164

OFFICERS AND COUNCIL 1938

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

51 Madison Ave., New York, N. Y.

President.....E. HOLT GURNEY

First Vice-President.....J. F. McINTIRE

Second Vice-President.....F. E. GIESECKE

Treasurer.....A. J. OFFNER

Secretary.....A. V. HUTCHINSON

COUNCIL

E. HOLT GURNEY, *Chairman*

J. F. McINTIRE, *Vice-Chairman*

Three Years

N. D. ADAMS — A. P. KRATZ, G. L. WIGGS — J. H. WALKER

Two Years

J. J. AEBERLY — M. C. BEMAN, W. A. RUSSELL — E. O. EASTWOOD

One Year

R. C. BOLSINGER, W. L. FLEISHER — D. S. BOYDEN — S. H. DOWNS, C. M. HUMPHREYS

COMMITTEES OF THE COUNCIL

Executive: D. S. BOYDEN, *Chairman*

W. L. FLEISHER, F. E. GIESECKE

Finance: J. F. McINTIRE, *Chairman*

S. H. DOWNS, E. O. EASTWOOD

Meetings: J. J. AEBERLY, *Chairman*

N. D. ADAMS, A. P. KRATZ

Membership: R. C. BOLSINGER, *Chairman*

W. A. RUSSELL, G. L. WIGGS

Standards Committee: C. M. HUMPHREYS, *Chairman*

M. C. BEMAN, J. H. WALKER

ADVISORY COUNCIL

D. S. Boyden, *Chairman;* Homer Addams, R. P. Bolton, W. H. Carrier, S. E. Dibble, W. H. Driscoll, H. P. Gant, John F. Hale, L. A. Harding, H. M. Hart, C. V. Haynes, E. Vernon Hill, J. D. Hoffman, John Howatt, W. T. Jones, D. D. Kimball, G. L. Larson, S. R. Lewis, Thornton Lewis, J. I. Lyle, F. B. Rowley, F. R. Still and A. C. Willard.

COOPERATING COMMITTEE

A. S. H. V. E. Representative on *A. S. A. Sectional Committee on Standardization of a Scheme for Identification of Piping Systems (A-13):* E. E. Ashley.

SPECIAL COMMITTEES

Committee on Admission and Advancement: T. H. Urdahl, *Chairman (one year);* M. F. Blankin *(two years),* and A. A. Adler *(three years).*

Publication Committee: F. C. McIntosh, *Chairman (one year);* W. M. Sawdon *(two years),* and W. H. Severns *(three years).*

Guide Publication Committee: Albert Buenger, *Chairman;* S. H. Downs, C. H. B. Hotchkiss, S. S. Sanford, G. L. Tuve, and E. N. McDonnell, *Advisory.*

Committee on Constitution and By-Laws: John Howatt, *Chairman;* O. W. Ott, and W. E. Stark.

F. Paul Anderson Award Committee: J. F. McIntire, *Chairman;* E. O. Eastwood, W. T. Jones, A. P. Kratz, and G. L. Larson.

Speakers Bureau Committee: W. A. Russell, *Chairman;* L. T. Avery, B. C. Candee, O. J. Prentice, and B. M. Woods.

NOMINATING COMMITTEE

Chapter	*Representative*	*Alternate*
Atlanta	E. W. Klein	C. T. Baker
Cincinnati	I. B. Helburn	H. E. Sproull
Golden Gate	B. M. Woods	
Illinois	J. R. Vernon	Tom Brown
Iowa-Nebraska	A. L. Walters	M. J. Stevenson
Kansas City	A. H. Sluss	A. L. Maillard
Manitoba	William Glass	R. L. Kent
Massachusetts	C. P. Yaglou	W. T. Jones
Michigan	F. J. Feely	F. J. Linsenmeyer
Western Michigan	W. W. Bradfield	L. G. Miller
Minnesota	R. E. Backstrom	M. S. Wunderlich
Montreal	L. H. Laffoley	G. L. Ballantyne
New York	W. E. Heibel	E. J. Ritchie
Western New York	B. C. Candee	J. J. Landers
Northern Ohio	L. T. Avery	G. L. Tuve
Oklahoma	E. F. Dawson	E. W. Gray
Ontario	H. R. Roth	C. Tasker
Pacific Northwest	C. W. May	L. Bouillon
Philadelphia	H. H. Erickson	H. H. Mather
Pittsburgh	J. F. Collins, Jr.	R. A. Miller
St. Louis	G. W. F. Myers	R. J. Tenkonohy
Southern California	E. H. Kendall	R. L. Gifford
Texas	W. H. Badgett	I. E. Rowe
Washington, D. C.	M. D. Kiczales	T. H. Urdahl
Wisconsin	C. H. Randolph	E. Szekely

[The material in this Section is prepared by the Publication Committee of the Society and nothing that appears in it may be reprinted, either wholly or in part, without permission. The use of the Society's name for advertising promotions is never authorized.]

Performance Tests of Asbestos Insulating Air Duct

By R. H. Heilman* (*MEMBER*), Pittsburgh, Pa. and R. A. MacArthur** (*NON-MEMBER*), Lockland, O.

THE all-asbestos duct is a new development in the air conditioning field although asbestos insulation has been used extensively on metal ducts. Data are presented briefly in this paper to indicate the suitability of asbestos for the fabrication of air conditioning ducts of satisfactory engineering characteristics. Friction losses, acoustic characteristics, and thermal data are presented and discussed.

Description of Asbestos Ducts

Asbestos ducts are of double-shell construction and at present are available in lengths of 3 ft. The stiff inner shell is approximately $\frac{3}{16}$ in. in thickness and is made from hardened laminated asbestos board. The outer shell, or jacket, makes a close sliding fit with the inner shell, or core, and consists of layers of corrugated or indented asbestos paper, having thermal insulating value, lined and covered with protecting hardened laminated asbestos paper. The standard overall thicknesses are ½ in. and 1 in.

Unretouched photographs of two sections of standard thicknesses are shown in Fig. 1. The inner shells have been partially withdrawn from the outer shell jackets.

For assembly of units into ducts, an inner shell is withdrawn as shown in Fig. 1 for a distance of, conveniently, 8 in. The extended core is telescoped into the jacket of another unit. This is continued until the required length is obtained. Adhesive tape or metal bands may be applied to the joints where the duct is exposed to view as the joint cracks would give an unsatisfactory appearance. A carpenter's saw is used for cutting cores or shells for fitting. The stiffness of these ducts is greater than that of sheet metal of usual gages, and the joints retain their integrity under considerable abuse. Ells and turns are assembled with asbestos directing vanes held in position with metal clips of special design. Figs. 2 and 3 indicate the method of fabrication of 90 deg ells. Tees are assembled and branches are taken from trunk ducts by combinations of straight lengths and turns, as indicated in Fig. 4. Sizes of ducts are established so that transformations, as indicated in Fig. 4, are possible over a wide range of size combinations.

Apparatus and Tests

Most of the data are presented from the practical viewpoint. For the most part, friction losses were directly measured for various air velocities and sizes of ducts by exploration of duct cross-sections with a portable air meter having a sensitivity of 0.005 in. A few static pressure drops were obtained for various lengths of ducts by means of water-filled manometers reading plenum chamber static pressures, velocities being obtained in the ducts by means of a pitot head and gage.

An attempt was made to determine the practical value of asbestos ducts in the reduction of sound in installations made therefrom. The sound measurement system in general consisted of (1) a sound-proofed room containing a centrifugal fan, a 3 hp motor and a sheet metal plenum chamber, (2) a second sound-proofed room lined with a 1 in. thickness of hair felt, and (3) a straight sound-proofed tunnel 76 ft in length connecting the two rooms. This tunnel contained the ducts to be tested. The walls of the rooms and tunnel were of double frame and plastered cane fiber insulation board construction. The rooms were insulated from the floor by layers of

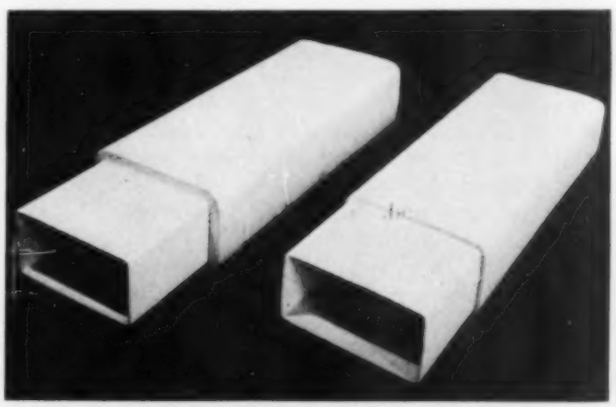


Fig. 1—Asbestos duct with inner shell partially withdrawn

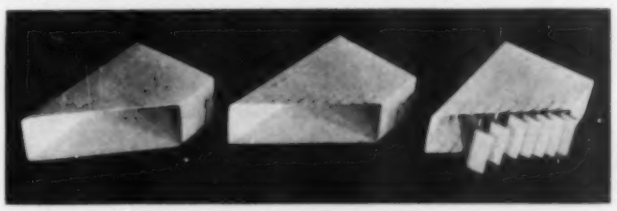


Fig. 2—Fabrication of 90 deg ell with asbestos directing vanes

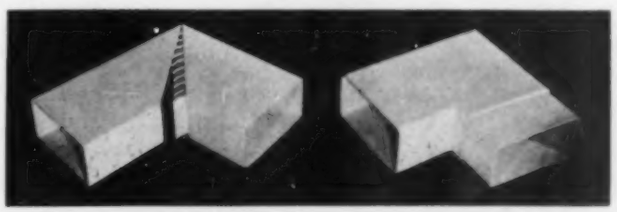


Fig. 3—Assembly of 90 deg ell with directing vanes

*Senior Industrial Fellow, Mellon Institute of Industrial Research.
**Research Chemist, The Philip Carey Manufacturing Co.
Presented at the 44th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at the Joint Session with the A.S.R.E., New York, N. Y., January, 1938

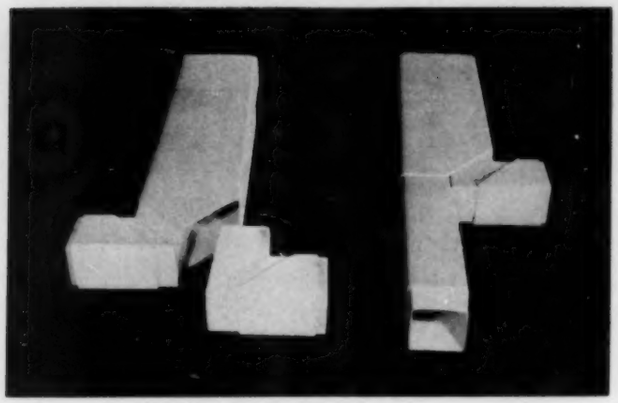


Fig. 4—Arrangement of asbestos air duct with tee and branch connections

sponge rubber and hair felt, and the tunnel was supported on strips of the same materials resting on wood supports. The fan, motor and plenum chamber were likewise supported from the floor of the room, as were ducts placed in the tunnel. As far as possible, sound travel from room to room other than through the duct was prevented. The sound intensity drop from the surroundings into this system was of the order of 55 db (decibels).

Sound of various constant frequencies and intensities was generated by means of a vacuum tube oscillator feeding an amplifier and speaker. Sound levels were measured with a sound meter reading decibels above the standard reference level of 10^{-16} watts per square centimeter.

For measurement of temperature drops through various lengths of ducts, gas-heated air was fed through a centrifugal fan and plenum chamber to the ducts. Temperatures and velocities were measured at selected points along the ducts under equilibrium conditions.

Friction Losses

Most engineers will primarily be interested in a comparison of the friction losses in the asbestos and metal ducts.

To obtain this comparison, the drop in static pressure for a distance of 57 ft was obtained on 11 in. x 13 in., 5½ in. x 8½ in., 2½ in. x 11 in. rectangular and on 8 in. round asbestos and metal ducts.

Fig. 5 gives the drop in static pressure obtained on the various ducts tested. The static pressure drop corrected to a 50 ft length is plotted against the average velocity at the downstream end of the duct.

The actual coefficient of friction of smooth metal pipes has been determined by various investigators. Emswiler,[1] for instance, found that the coefficient varied from about 0.008 to 0.004 for velocities of 200 to 2800 fpm. His experiments confirmed those of other investigators in the fact that the coefficient decreases with both increase of diameter and velocity. Later investigators have found that the coefficient of friction increases with increase of viscosity; however, the effect of the viscosity of the air is small and is usually neglected. An average value of $f = 0.0054$ is quite often used.

[1]Coefficient of Friction of Air Flowing in Round Galvanized Iron Ducts, by J. E. Emswiler (A.S.H.V.E. TRANSACTIONS, Vol. 22, 1916, p. 537).

The coefficient of friction for the asbestos and metal ducts was calculated from the test results by means of the relation:

$$P = \frac{fRL}{a}\frac{DV^2}{2g} \text{ or } h = \frac{fRL}{a}\frac{V^2}{2g} \qquad (1)$$

in which P = pressure required to overcome friction, pounds per square foot,
a = cross-sectional area of duct, square feet.
D = density of air, pounds per cubic foot.
V = velocity, feet per second.
f = coefficient of friction.
h = height, feet of a column of air equivalent to P.
R = perimeter of round duct, feet.
L = length of duct, feet.
g = acceleration due to gravity in feet per second per second,
or, in terms of inches of water, and for air at 80 F.

$$h = 0.00022\frac{fRL}{a}V^2 \text{ or } f = \frac{ha}{0.00022\,RLV^2} \qquad (2)$$

Since these formulae are for round ducts, the equivalent diameter of round pipe for equal friction per foot of length to carry the same capacity as the rectangular ducts was calculated from the relation:

$$d = 1.265\sqrt[8]{\frac{(ab)^5}{a+b}} \qquad (3)$$

in which a = one side of rectangular duct.
b = other side of rectangular duct.
d = equivalent diameter of round pipe for equal friction per foot of length to carry the same capacity.

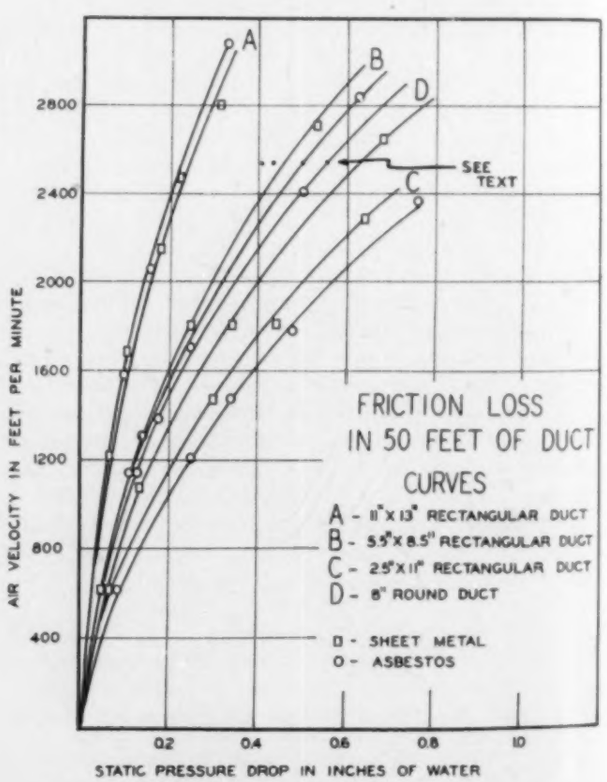


Fig. 5—Friction loss in 50 ft of asbestos and metal ducts

From formula (3), the equivalent diameters for the 11 in. x 13 in., the 5½ in. x 8½ in. and the 2½ in. x 11 in. ducts are 13.1 in., 7.49 in. and 5.5 in. respectively.

These equivalent diameters were used in calculating the coefficient of friction for the eight ducts. The static pressure drops for velocities of 1000, 2000 and 3000 fpm were taken from Fig. 5 and substituted in Equation (2). The results of the calculations are given in Table 1.

THE A.S.H.V.E. GUIDE has included for several years a chart for obtaining the friction loss in metal ducts. This chart is based on higher frictional resistances than usually occur in practice and, therefore, gives larger duct sizes so as to allow for a factor of safety for increased roughness of duct surface or for obstructions, etc.

The chart was calculated from the formula:

$$h_L = \frac{1.1\,L}{CD^{2/7}} \left(\frac{V}{4005}\right)^{19/7} \qquad (4)$$

in which h_L = loss of head, inches of water.
L = length of pipe, feet.
V = velocity of air, feet per minute.
D = diameter of pipe, feet.
C = length of pipe in diameters for one head loss.
C = 60 for perfectly smooth pipe; = 55 for pipe used in planing mill exhaust systems, and 50 for heating and ventilating ducts.

Since formula (4), using a value for $C = 50$, is used so extensively in designing air conditioning ducts, it is desirable that C values be calculated from the tests conducted on the 8 ducts. These values based on equivalent diameters of 13.1 in., 8.0 in., 7.49 in. and 5.5 in. are given in Table 1.

The static pressure drop for the same equivalent diameters as obtained from the chart in THE A.S.H.V.E. GUIDE, corrected for equivalent velocities, is included in Table 1 for comparison.

Referring to Table 1 it is seen that the values of C for the 8-in. round asbestos duct follow exactly the 1/7 power law of change in friction with change in velocity. The round metal duct is also fairly close to this relationship, and in general the rectangular asbestos and metal ducts follow closely this law for velocities of 2000 and 3000 fpm. The pressure drop is somewhat higher than this law requires for 1000 fpm in all rectangular ducts with the exception of the 2½ in. x 11 in. ducts. For these ducts the relationship is reversed and since the metal duct shows so much lower drop in relation to the other sizes, it is planned to continue the tests on the 2½ in. x 11 in. metal duct.

The variation of friction with pipe diameter in these tests did not check the 2/7 power law of change in friction with diameter of duct as closely as the check on the velocity law. However, in all cases with the exception of the 8-in. round ducts the coefficient of friction decreased with increase in equivalent diameter of duct.

An inspection of the values of C indicates that in all cases these values are considerably higher than the value of 50 recommended in THE A.S.H.V.E. GUIDE for air conditioning systems. It is noted that C for the 8-in. round metal duct is roughly 20 per cent higher than the recommended value. However, the coefficient of friction obtained for both round ducts was considerably higher than that obtained on the rectangular ducts. The higher values of C obtained on the rectangular ducts indicate that it is inaccurate to determine friction losses for rectangular ducts from the A.S.H.V.E. chart which is based on test results obtained on round ducts.

It is believed that the use of a value of 50 for C results in ducts considerably larger than is actually necessary for good engineering practice.

If one totals the drop in friction for all sizes and all velocities observed for the asbestos and the metal ducts it will be found that there is less than 1 per cent difference in pressure drop between the two types of ducts. This would indicate that there is probably no practical difference in friction between the asbestos and metal ducts.

A comparison of the friction losses through asbestos and metal fittings has been made on several sizes of 90-deg elbows, and it is assumed that data on these fittings are applicable, for purposes of comparison, to other fittings.

Fig. 6 shows the experimental values of plenum chamber static pressure necessary to force air, at the velocities given, through 42 ft of 5½ in. x 8½ in. asbestos duct equipped (1), with a metal 90 deg broadway sweep turn of radius ratio 1.5 and (2), with a 90-deg asbestos square turn equipped with asbestos turning vanes. Within the limits of experimental error, the loss is the same for both turns, and under experimental conditions, the loss through each was about 22 per cent of the velocity head for all velocities.

Table 1—Results of Friction Tests on Asbestos and Metal Ducts

Velocity, Feet per Minute	Static Pressure Drop, Inches of Water per 100 Ft			Coefficient of Friction f		Value of C	
	Asbestos	Metal					
	Observed	Observed	A.S.H.V.E.	Asbestos	Metal	Asbestos	Metal
11 in. x 13 in. Rectangular Ducts							
1000	0.098	0.098	0.165	0.00391	0.00391	84.5	84.5
2000	0.310	0.310	0.600	0.00308	0.00308	97.0	97.0
3000	0.645	0.670	1.270	0.00286	0.00297	98.3	94.8
8 in. Round Ducts							
1000	0.200	0.235	0.280	0.00543	0.00638	70.0	59.5
2000	0.725	0.825	1.020	0.00492	0.00560	70.0	61.7
3000	1.545	1.735	2.150	0.00466	0.00522	70.0	62.5
5½ in. x 8½ in. Rectangular Ducts							
1000	0.200	0.195	0.340	0.00420	0.00443	85.0	87.2
2000	0.665	0.595	1.270	0.00377	0.00333	92.5	103.5
3000	1.395	1.300	2.600	0.00353	0.00328	93.5	100.0
2½ in. x 11 in. Rectangular Ducts							
1000	0.341	0.267	0.600	0.00474	0.00371	93.2	118.5
2000	1.175	1.032	2.500	0.00410	0.00360	89.8	102.0

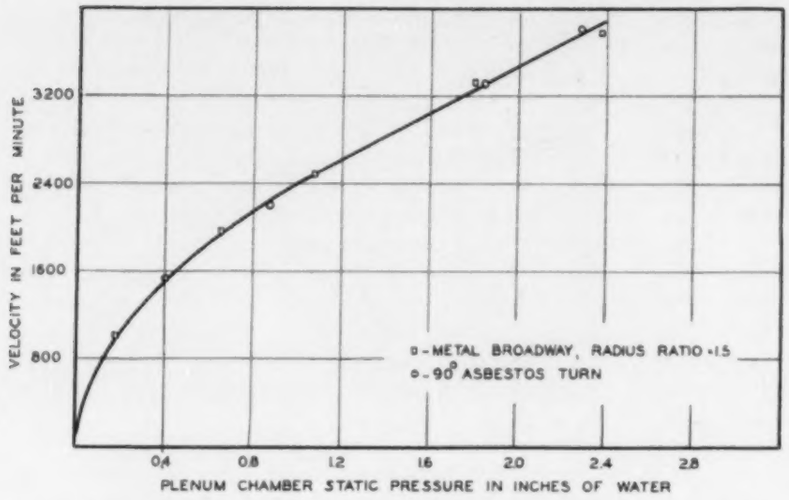


Fig. 6—Plenum chamber static pressure to force air through 42 ft of 5½ in. x 8½ in. duct including 90 deg turn

Published values for the friction losses through metal elbows of square cross-section and radius ratio equal to 1.5 vary from 9 per cent to 40 per cent of the velocity head,[2] depending upon the conditions. The values plotted in Fig. 6 were obtained with 24 ft of duct between the plenum chamber and the turn, and the turn was followed by 18 ft of additional duct.

With the metal turn under test, the velocity distribution was not uniform across the end of the 18 ft leg after a distance corresponding to nearly 30 diameters of duct. Any evaluation of the losses in turns, or in ducts, for that matter, should be accompanied by the conditions of test. Air straighteners are commonly used in tests of this nature. Straighteners are not commonly used in practice, and tests made therewith should not be used to compute losses in practice. The test shown in Fig. 6 was merely comparative, and no attempt was made to measure other than plenum chamber pressures and average duct velocities. The entrance loss to the duct was known to be very high. Values obtained for the loss in identical turns varied from 85 per cent of the velocity head to 17 per cent, depending entirely on conditions.

Points indicated on Fig. 5 by the arrow show observed values of the friction loss in 5½ in. x 8½ in. asbestos ducts; the apparent friction loss per unit length decreased markedly with increase in length of tested duct. No apology is offered for the method of test, as the systems used simulate practical installations, and all curves in Fig. 5 were obtained under comparable conditions.

Heat Transmission Characteristics

Two conductivity tests were run on ½-in. thick asbestos duct samples and two conductivity tests were run on 1-in. thick asbestos duct samples. Since there was no appreciable difference in the conductivity (per inch of thickness) of the 1-in. and ½-in. thick samples the average conductivity of the four tests only is given. The thermal *conductance* of the 1-in. thick duct is of course one-half that of the ½-in. thick duct. The conductivity as measured in these tests was 0.622 Btu at a mean temperature of 50 F and 0.707 Btu at a mean temperature of 150 F. The conductivity at mean temperatures between these two values is a straight line function of mean temperature.

The thermal transmission coefficient U for various temperature conditions, etc., can be calculated from the standard formula,

$$U = \frac{1}{\frac{1}{f_i} + \frac{L}{K} + \frac{1}{f_o}} \qquad (5)$$

where

U = Btu per square foot per hour per degree Fahrenheit difference in temperature between the air inside and the air outside the duct.

f_i = film conductance inside the duct, Btu per hour per square foot per degree Fahrenheit.

f_o = film conductance outside the duct, Btu per hour per square foot per degree Fahrenheit.

K = conductivity of the duct, Btu per hour per square foot per degree Fahrenheit per inch.

L = thickness of the duct, inches.

It will be found that f_i will vary considerably with the velocity and the size of the duct. THE A.S.H.V.E. GUIDE gives a value of $f_i = 6$ for outside walls and for a wind velocity of 15 mph. Various investigators show lower values than this for ducts 6 in. in diameter or over. For the purpose of making approximate calculations an average value of $f_i = 4$ has been used in this paper.

The outside surface conductance f_o will vary considerably with the size of the duct for small ducts and for metal ducts it will vary considerably with the emissivity p of the duct. Values of f_o for various conditions can be calculated from data given by Heilman[3] and others. Previous tests were conducted on galvanized metal to determine the emissivity of bright and tarnished material. Very bright galvanized metal had an emissivity of 0.14, while badly tarnished metal had an emissivity of 0.53. Horizontal metal ducts will ordinarily be covered with a layer of dust on the top surfaces, so that part of the duct will have a very high emissivity value. Some metal ducts are covered with asbestos paper. In this case the emissivity is 0.93.

Tests were conducted on various sizes of ½-in. thick asbestos ducts and on an 11 in. x 13 in. metal duct insulated with ½ in. thickness of cellular laminated asbestos paper insulation to determine the drop in temperature in 30 ft of duct. These tests were run with velocities of approximately 650 to 1750 fpm and with initial air temperatures of approximately 160-230 F.

THE A.S.H.V.E. GUIDE 1937, page 176, gives a formula for calculating the temperature drop in ducts. For low velocities and long ducts of small cross-section a somewhat more accurate formula may be used as follows:

[2]Pressure Losses in Rectangular Elbows, by R. D. Madison and J. R. Parker (*Heating, Piping and Air Conditioning*, July, August, September, 1936).

[3]Heat Insulation in Air Conditioning, by R. H. Heilman (*Industrial and Engineering Chemistry*, Vol. 28, July, 1936, p. 782).

Table 2—**Thermal Transmission Coefficients for Metal and Asbestos Ducts**

Ducts	Btu per Hour per Square Foot per Degree Fahrenheit, U
½ In. Thick Asbestos, Calculated	0.662
½ In. Thick Asbestos, Observed	0.668
Bright Metal	0.832
Tarnished Metal	1.11
Metal, Covered with Asbestos Paper	1.33
Metal, Insulated with ½ In. Thick Cellular Laminated Asbestos Paper, Calculated	0.482
Metal, Insulated with ½ In. Thick Cellular Laminated Asbestos Paper, Observed	0.523

Table 3—**Heat Transmission Values for Metal and Asbestos Ducts**

Ducts	Heat Transmission, Btu per Square Foot per Hour
½ In. Thick Asbestos	12.1
Tarnished Metal	17.5
Metal, Covered with Asbestos Paper	23
Metal, Covered with ½ In. Thick Cellular Laminated Asbestos Paper	8.8

$$T = \frac{T_i - T_o}{\left(e^{\frac{UPL}{60Advc}}\right)} + T_o \qquad (6)$$

where

T = final temperature of air in duct, degrees Fahrenheit.
T_i = initial temperature of air in duct, degrees Fahrenheit.
T_o = outside air temperature, degrees Fahrenheit.
U = thermal transmittance of duct, Btu per hour per square foot per degree Fahrenheit.
P = perimeter of duct, feet.
L = length of duct, feet.
A = area of duct, square feet.
d = density of air, pounds per cubic foot.
v = velocity of air, feet per minute.
c = specific heat of air, Btu per pound per degree Fahrenheit.
e = Naperian base of logarithms = 2.718.

Equation (6) can also be used in calculating the thermal transmittance U of a duct when the drop in temperature through a given length of duct is known or measured. The thermal transmittance for the various sizes of asbestos ducts on which temperature drop determinations were made and also for the insulated metal duct have been calculated from equation (6) and the observed temperature drops. The average transmittances for three sizes of asbestos ducts with an average internal temperature of 166 F and an outside temperature of 85 F and the transmittance for the metal duct insulated with ½-in. thick cellular laminated asbestos paper for the same temperature conditions are given in Table 2.

Thermal transmittance values have also been calculated for various types of ducts from equation (5) and the known values of K as determined for asbestos ducts and cellular laminated asbestos paper insulation. In calculating the transmittance values for the bright, the tarnished, and the asbestos paper-covered metal ducts, the emissivity values used were 0.14, 0.53 and 0.93, respectively. In these calculations an average air temperature of 166 F in the duct and an outside air temperature of 85 F were used. A value of 4 was used for f_1. The results of these calculations are as given in Table 2.

The heat transfer from metal and asbestos ducts is lower for air cooling installations owing to the lower temperatures and to lower temperature differences.

The heat transmission in Btu per square foot per hour for various types of ducts as calculated from equation (5) is given in Table 3. These calculations were based on an average temperature of 60 F in the duct with an outside air temperature of 80 F. The emissivity values and the film conductances which were used in the previous calculations were also used in these calculations.

The drops in temperature through 50 ft of various sizes of ½-in. thick asbestos ducts, tarnished metal ducts, and metal ducts insulated with ½-in. thick cellular laminated asbestos paper have been calculated by means of equation (6) using coefficient of transmission values of 0.662, 1.11 and 0.482 respectively. In these calculations an outside air temperature of 70 F was assumed. The calculated values are given in Table 4.

Sound Characteristics

Various tests of a practical nature were made on the behavior of galvanized sheet metal ducts and asbestos ducts with respect to sound transmission.

In general, intensities of sound generated in a plenum chamber by means of an oscillator of variable frequency feeding a speaker at variable intensities, or by means of a fan, blowing air through the system, were measured at both ends of ducts of various sizes but of fixed length of 76 ft. The system was sufficiently well insulated that extraneous noise effects as well as transmission of noise from the sources used other than through the ducts under test affected the measured levels to an unmeasurable degree.

The oscillator frequencies available were limited to 100, 250, 500, 1000 and 2000 cycles per second. Obtaining definite values of the actual transmission of sound through ducts under these conditions is very difficult, as has been indicated in the literature on sound measurement. The chief difficulty in measurement of levels lies in the variable sound intensity pattern, which shifts with each frequency used and which, also, prevents exact exploration of duct openings other than mechanically, as the observer's body has marked effect on measured intensity levels.

The data obtained are of such nature that only indications may be mentioned. As is to be expected, asbestos

Table 4—**Calculated Temperature Drop Through 50 Ft of Duct Initially at 175 F, Outside Air Temperature, 70 F**

Duct Size	Velocity 500 fpm			Velocity 1000 fpm		
	Asbestos Duct	Tarnished Metal	Insulated Metal	Asbestos Duct	Tarnished Metal	Insulated Metal
2½ in. x 11½ in.	59.5	79.1	48.0	36.0	53.0	27.6
5½ in. x 11½ in.	38.8	66.2	38.0	21.7	34.2	16.2
11½ in. x 11½ in.	27.1	41.5	20.6	14.7	23.3	10.7
11½ in. x 23½ in.	21.0	32.7	15.7	11.5	17.9	8.1
23½ in. x 23½ in.	14.4	22.8	10.8	7.4	12.3	5.5

ducts transmit less sound than metal ducts, the difference increasing with frequency.

Due presumably to the large number of component frequencies of the sound developed (eliminating a fixed intensity pattern in the rooms), the noise levels caused by the fan and the air it fed through the duct were measured with relative ease and the values obtained were reasonably reproduced.

In Fig. 7 are shown curves comparing the sound intensities in the room at the end of the ducts indicated and the intensity drops through the ducts at the given air velocities. The measured sound level in the plenum chamber was about 90 db above the reference level of 10^{-16} watts per square centimeter. The level in the plenum chamber at duct air velocities of 3500 fpm was about 2 db higher than at a velocity of 500 fpm. That this difference is no greater is probably due to the fact that the air velocity was varied by means of a damper at the fan inlet; the fan noise should be about the same for widely different velocities. In practice, a smaller fan would be used for lower velocities; for this reason the plenum chamber sound levels at lower velocities probably would be lower than those given. All data in Fig. 7 are referred to a plenum chamber noise level of 90 db.

The drops in sound intensity through asbestos ducts of all sizes are higher than those through the corresponding metal ducts, and the improvement is substantially constant up to velocities of 3500 fpm. At a velocity of 1000 fpm, the sound levels in the room at the exhaust end of the ducts were lower with asbestos ducts by the following amounts: 2½ in. x 11 in. ducts, 8.8 db; 5½ in. x 8½ in. ducts, 5.7 db; 11 in. x 13 in. ducts, 6.0 db.

It should be noted that the levels given in Fig. 7 are higher than ordinarily obtained; levels were kept high to facilitate accurate measurement under variable conditions, and the levels were obtained close to the duct outlets. The ducts tested were straight, with no fittings, turns or other sound absorbing equipment which would have made the levels lower in an actual installation. Plotted levels at this point are the averages along fixed points in the room, and all data were obtained under standardized conditions. The effect of the standing wave pattern and of the observer's body are negligible when making observations on sound of this nature.

For the intensity levels given in Fig. 7, the average apparent loudness of the noise issuing from the asbestos ducts is as follows:[4]

Velocity, 1000 Fpm

2½ in. x 11 in. asbestos duct, loudness approximately 40 per cent that of metal.

5½ in. x 8½ in. asbestos duct, loudness approximately 60 per cent that of metal.

11 in. x 13 in. asbestos duct, loudness approximately 65 per cent that of metal.

It has been noted that the sound issuing from the metal ducts under test contained equipment noise to a variable extent. With the asbestos ducts equipment noise apparently is absent, under comparable conditions.

The data of Fig. 7 also indicate that air velocities in asbestos ducts may be increased greatly over present customary velocities without exceeding the present sound intensities at the duct outlets.

Data obtained using fixed frequencies check within the limits of error the data obtained with air. However, the data at low frequencies indicate that for small ducts, as tested here, the entrance and exit drops in sound level predicted by the theory of sound transmission through conduits of cross-section much less than one wave length of sound are of considerably more importance, and drops obtained at low frequencies are higher than would be expected from the data obtained at higher frequencies.

The sound level drop through a 5½ in. x 8½ in., 90 deg asbestos turn, equipped with asbestos turning vanes, was compared with that through a metal sweep turn of the same cross-section and of radius ratio 1.5, both with sound of fixed frequency and with air. The asbestos turn decreased sound of frequencies 250, 500 and 1000 cycles per second approximately 7, 11 and 14 db, respectively, more than the metal turn, which had virtually no effect. With air the differences were approximately 4 and 7 db for velocities of 500 and 1000 fpm. There was no measurable difference between the two turns at 2000 and 3000 fpm. In a system containing several asbestos fittings, a much lower exit sound level may be expected at ordinary velocities than in systems that contain only metal sweep turns.

Sound is picked up by asbestos ducts less readily than by bare metal ducts, the effect increasing considerably with frequency of the sound and thickness of the duct. The sound picked up by a duct, independently of that by conduction, should be approximately inversely proportional to the weight per square foot of area.[5] The value of a duct of this nature lies in its ability to absorb picked up sound. As is to be expected, there is

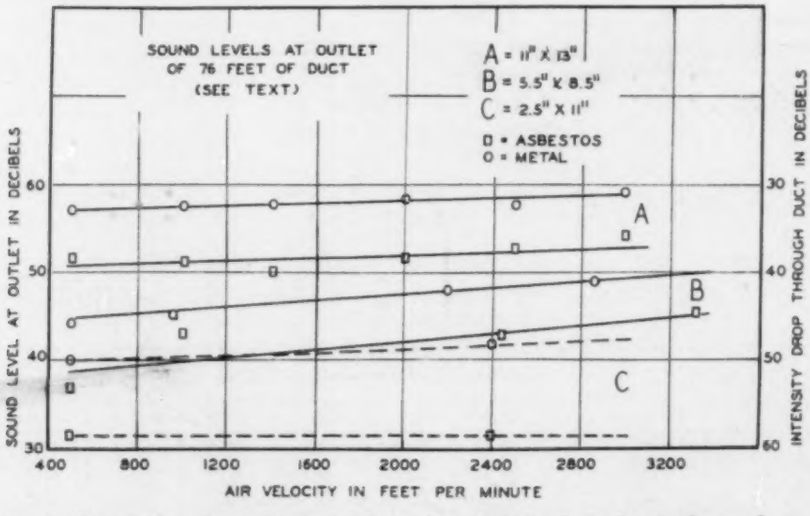


Fig. 7—Comparison of sound intensity drops through 76 ft of metal and asbestos ducts

[4]American Tentative Standards for Noise Measurement (*American Standards Association*, Feb. 17, 1936).

[5]Acoustics and Architecture, by Paul E. Sabine (McGraw-Hill, 1935, p. 279).

Acoustics, by Geo. W. Stewart and R. B. Lindsay (D. Van Nostrand and Co., 1930, p. 130).

no sounding board effect on asbestos duct with changes in temperature, with opening of dampers, etc., and the sound of dampers and equipment noises are considerably diminished over those connected to metal ducts. There is no measurable sound conduction through the material itself. The sound intensity measured at 3 ft from the side of a metal duct at the end of which a damper was opened and closed, starting and stopping air flow, was about 90 db. This is probably higher than is ordinarily the case with dampers.

Conclusions

The results of this preliminary investigation indicate that the asbestos ducts have practically the same coefficient of friction as metal ducts. The tests also indicate that the Friction Chart given in THE A.S.H.V.E. GUIDE requires considerably larger ducts than is necessary for good engineering practice.

Conductivity data on the asbestos duct are given. Also, calculated and measured temperature drops for various ducts are included.

The tests indicate that present usual air velocities can be increased many times without undue noise in the asbestos ducts. For all air velocities tested, the asbestos ducts were superior to metal ducts in sound attenuation.

A. K. Purdy Dies Suddenly

New Year's Day was saddened in Toronto with the news of the sudden death of A. Kirk Purdy, which occurred that morning following a heart attack.

Born in Toronto 44 years ago, Mr. Purdy was educated at local schools and the University of Toronto. He served over-seas with the Canadian Engineers' Corps. Upon leaving the University, Mr. Purdy entered the firm of Purdy-Mansell, Ltd., with which he has been associated all his business life, with the exception of one year served with General Fire Extinguisher Co., Ltd., Montreal. Following the death of his father in 1922, Mr. Purdy was made president of Purdy-Mansell, Ltd.

Mr. Purdy became a Member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in 1922 and he belonged to the Ontario Chapter. He was an active member of the Toronto Builders' Exchange and of the *Canadian Construction Association*. At one time he served as chairman of the Plumbing and Heating Section of the Exchange. At the time of his death, he was chairman of that section's Labor Relations Committee, and also a member of the Advisory Committee for the plumbing trade under the Industrial Standards Act. A former chairman of the Trade Contractor's Section of the *Canadian Construction Association*, he was also a former honorary treasurer of that body and was well known to the membership at large. He was a member of the Granite Club, the Zetland Masonic Lodge, St. Patrick's Chapter, Cyrene Preceptory and Rameses Shrine.

Mr. Purdy is survived by his wife, the former Isabelle Saunders; two children, William and Judith; and two brothers, F. William and Dr. A. T. Purdy, both of Toronto.

Death of W. B. Clarkson

Friends of W. B. Clarkson, Owatonna, Minn., were saddened by his death, which occurred at the City Hospital, on December 23, following a ten-day critical illness resulting from injuries sustained in an automobile accident in Chicago on December 10, complicated by pneumonia. After receiving injuries, he continued a business trip to Quincy, Ill., and then drove home, collapsing as he arrived at his doorstep on December 14.

Mr. Clarkson, president of the King Ventilating Co., one of Owatonna's largest industrial concerns, was long a leader in Masonic and United Commercial Travelers' affairs, and served as grand counselor of the U. C. T.'s Minnesota-North Dakota-Saskatchewan jurisdiction in 1911-12.

Mr. Clarkson was born in Alton, Ill., July 12, 1862, and as a young man served in the U. S. Navy. After this service he went to Missouri and joined his brother in the implement business. He moved to Des Moines, Iowa, entering the sales field there, and then to Albert Lea, before coming to Owatonna in 1908. For five years he traveled as an implement salesman and in February, 1913, he became associated with the King Ventilating Co. as vice-president.

Since 1919 he had belonged to the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, of which he was a Life Member. He was a member of the *American Society of Refrigerating Engineers* and a former vice-president of the *American Society of Agricultural Engineers*. He was a veteran member of the Board of Directors of the *Minnesota Employers Association*. He was recognized by colleges and other scientific agencies as an authority on ventilating and air conditioning and was scheduled to address gatherings to be held at the Universities of Wisconsin, Missouri and Michigan shortly.

Surviving Mr. Clarkson are three daughters, Miss Bessie B. Clarkson, Dayton Bluff Station, Minn., Mrs. Robert D. Nelson and Mrs. J. J. Miller of Owatonna, and six grand children. A brother, J. D. Clarkson, West Plains, Mo., also survives.

A.S.A. Holds Annual Meeting

On December 1 the Annual Meeting of the *American Standards Association* was held at the Astor Hotel, New York, and the largest increase in membership of any year since the Association was formed was reported.

Officers elected for 1938 are D. D. Barnum, president, E. A. Prentis, vice-president and F. M. Farmer, chairman of the Standards Council, who were re-elected to offices which they held during the past year. R. P. Anderson was elected vice-chairman of the Standards Council.

During the past year 59 standards have been approved, among them being the Standard for Railway Grade Crossing Protection and the American Standard for Large Rivets. Numerous standards have been revised, among them being the Elevator Safety Code, the National Electrical Code, the American Standard for Compiling Industrial Injury Rates, and the Safety Code for Power Presses.

Effects of Artificial Lighting on Air Conditioning

By Walter Sturrock* (*NON-MEMBER*), Cleveland, Ohio

THE air conditioning of a building or any part of it for the summer months involves several problems, one of which is that of maintaining comfortable air temperatures. The discussion in this paper refers to the effect of artificial lighting on the temperature rise which must be considered in designing the cooling system. It is needless to state that natural light from the sun and sky coming through windows carries with it a considerable amount of radiant energy much of which becomes sensible heat when it strikes an absorbing surface. Likewise, artificial light sources emit heat, most of which is in the form of radiant energy, but becomes sensible heat as soon as it strikes an absorbing surface.

Reproduction of the so-called cold light, such as that made by the firefly, is known to be possible by mixing a luminol-caustic soda solution with a hydrogen peroxide-potassium ferricyanide solution. The cost, however, of producing cold light by this method is extremely high.

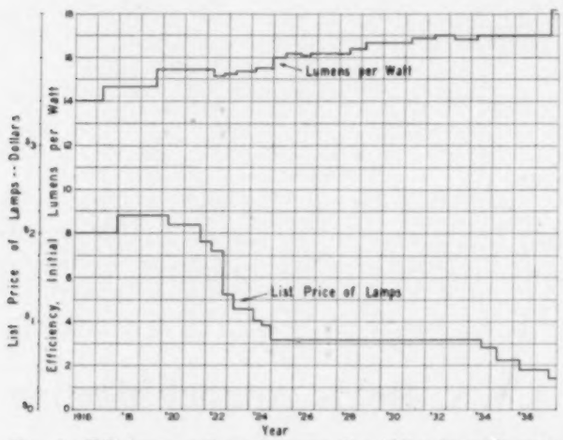


Fig. 1—Efficiency and cost curves for 200-w incandescent lamps

It is more than a million times as great as that with modern incandescent lamps, and therefore needs no further consideration here. All practical light sources generate heat as well as light, and there is a definite heat equivalent for each watthour of energy consumed, which amounts to 3.415 Btu. While this heat equivalent per watthour for electric lamps is a constant value, it is interesting to note that due to the many researches, principally during the past quarter century, the light units generated by each watt have gradually increased. For example, the 200-w (watt) gas-filled lamp, which has been on the market more than 20 years, appears today about the same as when first made available, but its light output has changed as follows: 20 years ago 1 kw of lighting load using five 200-w lamps produced about 14,000 lumens of light; 15 years ago it produced 15,100 lumens; 10 years ago 16,200; 5 years ago 16,900, and today it has increased to 18,200. This last increase in light output was made possible by the high efficiency 750-hour lamp which became practical by recent price reductions. The increases in lumens per watt over a 20-year period are shown graphically in Fig. 1, and also the reduction in lamp prices, both of which are important factors as far as the economics of artificial lighting is concerned.

The levels of illumination (footcandles) obtained from artificial lighting, in general, are so low in comparison to the thousands of footcandles found out-of-doors that the public has not only always been eager to accept increases in the efficiency of producing light in order to provide better illumination, but has in many instances demanded double or perhaps triple the connected wattage load used 10 to 15 years ago. During all these advances there is, of course, no change in the fact that each watt of lighting load continues to generate 3.415 Btu per hour.

Heating Effect From Lighting Installations

From the growing interest in adequate lighting in offices, stores, factories, and other work places, the illumination levels for the future will unquestionably be higher than those of the past. These increased levels will in many cases be obtained by simply using higher wattage lamps. In other cases, and particularly where air-conditioning is to be used, it will be desirable to revamp the entire lighting system or in new buildings to give special attention to the lighting in order to obtain a high utilization of light with its corresponding minimum wattage requirement. Some of the considerations along this line of development are discussed in this paper.

Higher Utilization of Light

The total wattage required to provide a specified level of illumination for an interior depends to a great extent on the percentage of the light generated by the lamp (utilization coefficient) which actually reaches the working area. This utilization of light depends not only upon the ability of the sidewalls and ceiling to redirect light but also upon the total light output of the reflector or diffusing globe in which the lamp is placed. Recent developments in higher reflection factors for paints and in the light reflection properties of aluminum are of special interest. A new finish, known as Alzak, has not only increased the light output of aluminum reflectors but also has provided a surface which resists abrasion and when dirty can be readily cleaned to restore its initial efficiency. A recent lighting installation for a small office,

*Illuminating Engr., Nela Park Engrg. Dept., General Electric Co.
Presented at the 44th Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., January, 1938.

Fig. 2—Parabolic sectional designed aluminum ceiling in office provides 33 per cent higher utilization of light than ordinary indirect lighting

shown in Fig. 2, was designed so that small indirect aluminum reflectors directed the light toward a special ceiling which was also of aluminum. In this case, the utilization of light was 32 per cent as compared with 24 per cent which would have been obtained from a conventional lighting system. In other words, in order to get an equal level of illumination from a conventional system the wattage load would have been 1⅓ times as great.

A large public utility located in Detroit has recently completed the lighting design for a new office building. A feature of particular interest is the use of ceiling coffers for lighting units. They are about 4 ft square and 15 in. deep and are designed to utilize the space between the large air ducts in the ceiling running lengthwise of the building. By using a silvered bowl lamp in each of these coffers the utilization of light is approximately 42 per cent, which is exceptionally good. An ordinary indirect system if installed would have had a utilization of not over 28 per cent, which would have required 50 per cent more watts to provide the 35 footcandles in service obtained from the adopted system. Fig. 3 shows a cross-section of a typical floor. Other examples of well-designed lighting systems could be discussed, but suffice it to say that more thought should be given to this phase of the problem especially for new or remodeled buildings where it is definitely known that an air cooling system will be installed.

Minimizing Heat by Reduction of Light

The quantity of heat accompanying light can always be reduced by making a corresponding reduction in the footcandles of illumination. In the case of natural light coming through the windows, awnings can be installed to intercept the radiant energy but likewise they intercept a high proportion of the light and are often objectionable for this reason. In artificial lighting, smaller sized lamps will reduce the heat as well as light, but this is objectionable from the standpoint of the footcandles required. If lower wattage lamps are used to reduce the heat during the summer months, when air cooling is desired, then in the same manner as awnings are stored away in the winter the lower wattage lamps should be replaced early in the fall by lamps of the proper size to provide adequate illumination. A replacement of lamps at different seasons of the year is not, however, in general a practical solution to the problem. For buildings having considerable window area for natural light a better solution is to arrange the wiring system so that the lamps near the windows can be turned off at times when daylight is sufficient in this area. During the summer months the turning off of lamps near the window in these buildings would not be serious. On the other hand, during the winter months artificial lighting is needed over the entire area to supplement daylight, particularly for the early forenoon and late afternoon hours.

Psychological Effects of Lighting

The yellowish appearance of some artificial light sources makes a room appear warmer. On the other hand, artificial light sources producing a whiter light make a room psychologically cooler even though lamps of a higher wattage may be used.

With the adoption of higher levels of illumination, particularly in smaller rooms, the color of the vertical surfaces in the line of vision has become a very important psychological factor. From this point of view the cooler colors in side wall finishes are desirable. In selecting

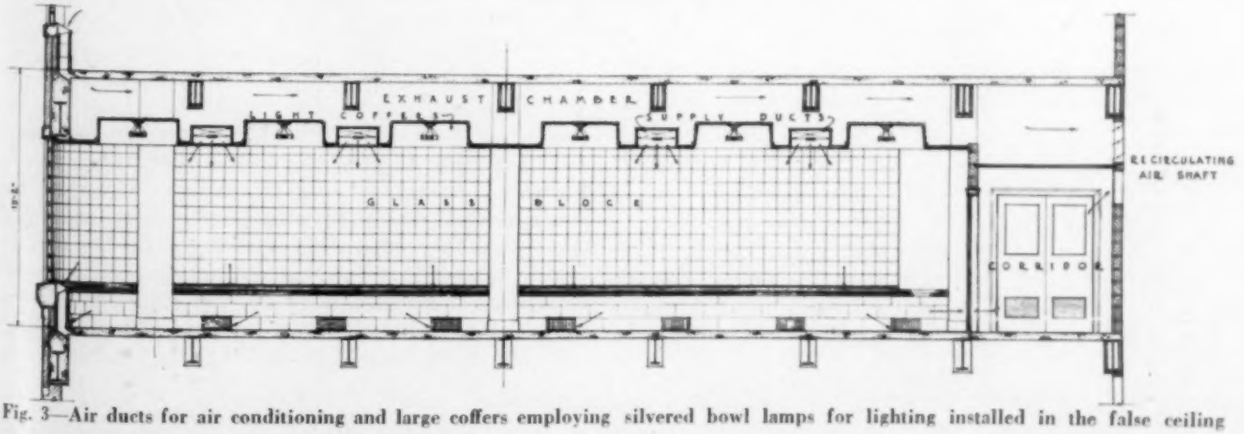


Fig. 3—Air ducts for air conditioning and large coffers employing silvered bowl lamps for lighting installed in the false ceiling

Table 1—A Group of Interiors with Recommended Minimum Standards of Illumination and the Approximate Lighting Requirement in Watts Per Square Foot of Floor Area

	Recommended Footcandles	Electrical Demand Watts per Square Foot
Auditorium	5	1.5
Automobile Show Room	20	4.0
Banks		
Lobby	10	2.5
Counters (75 w per running foot)	50-100	
Barber Shop and Beauty Parlor	20	4.0
Bowling		
Alley Runway and Seats	10	2.5
Pins (300 w per set)	30-50	
Billiards		
General	10	2.5
Tables (450 w per table)	30-50	
Club and Lodge Rooms		
Lounge and Reading	20	4.0
Auditorium	5	1.5
Court Rooms	10	2.5
Dance Halls (with no color or spectacular effects)	5	1.5
Drafting Rooms	30	7.0
Hotels		
Lobby (with no provision for convention exhibits)	10	2.5
Dining Room	5	2.5
Kitchen	10	2.5
Bedrooms (including allowance for table and bed lamps)	10	3.0
Corridors (10 w per running foot)	2	
Writing Room (including allowance for portable lamps)	20	4.0
Library		
Reading Rooms (including allowance for table lamps)	20	6.0
Stack Room (12 w per running foot of facing stacks)	10	
Moving Picture Theaters (with no color or spectacular effects)		
During Intermission	5	1.5
During Pictures	0.1	0.5
Museum		
General	10	2.5
Special Exhibits (an average of 2.5 w per square foot)	50-100	
Office Buildings		
Private and General Office		
No close work	10	3.0
Close work	20	5.0
Conference Rooms	10	3.0
Filing and Mail Sorting	20	5.0
Reception Rooms	10	3.0
Corridors and Stairways	5	1.5
Post Office		
Lobby	10	3.0
Sorting, Mailing, etc.	20	4.0
Office, Private and General	20	5.0
File Room and Vault	10	2.5
Corridors and Stairways	5	1.5
Professional Offices		
Waiting Rooms	10	2.5
Consultation Rooms	20	5.0
Operating Rooms	20	5.0
Dental Chairs (600 w per chair)	30-50	
Restaurants, Lunch Rooms and Cafeterias		
Dining Area	10	2.5
Food Displays (100 w per running foot of counter)	30-50	
Stores		
Main business districts—large cities	20	5.0
Neighborhood stores—large cities and all stores—small cities and towns	15	3.5
Show Cases		
40 w per running foot	50-100	
Show Windows		
Brightly Lighted Districts, large cities 300 w per running foot of glass	200	
Secondary business districts, large cities 150 w per running foot of glass	100	
Neighborhood stores, large cities and for small cities and towns—75 w per running foot of glass	50	
Lighting to reduce daylight reflections—500 w per running foot of glass	400	
Theaters		
Auditoriums (with no color or spectacular effects)	5	1.5
Foyer	10	2.5
Lobby	15	3.5

these cooler colors, it is not necessary to choose a side wall finish of extremely low reflection factor, which would not only be depressing but would also create an undesirable contrast. In general, side wall finishes having a reflection factor of 45 to 55 per cent are quite satisfactory. Colors such as the light greens, light blues and various mixtures of them come within this range. They have been satisfactorily used in many interiors where 30 or more footcandles of illumination were installed.

It is believed that psychological factors deserve careful consideration in connection with the advancement of air conditioning along with higher levels of general lighting.

Standards of Illumination

There is no constant ratio showing the relation between the level of illumination and the wattage consumed for lighting. Larger lamps are more efficient in light production than smaller ones. Furthermore, the light output and distribution efficiency of certain lighting units give them an outstanding advantage over other lighting units when installed for a particular application. Moreover, in some cases, double the wattage is required as in others in order to provide the same level of illumination. Therefore, the best procedure for determining the actual wattage consumed by the lighting system is to survey the installation and make a summation of the lamp sizes. In new or remodeled buildings the summation of lamp wattages should be obtained from approved and accepted lighting specifications. In any case it will be helpful to know the minimum footcandle values and the corresponding approximate watts required for modern lighting levels in various interiors in which air conditioning may be desired. Table 1 shows a list of interiors with the approximate lighting loads given either in watts per square foot of floor area, or in other terms as indicated. The recommended footcandles represent the order of magnitude rather than exact levels of illumination.

Energy Analysis from Incandescent Lamps

When a gas-filled incandescent lamp of the 300 to 500-w size is operated at normal voltage in still air its energy distribution is about as given:[1]

Radiation in the visible spectrum	11 per cent
Heat as invisible radiation in the infra-red region	70 per cent
Heat which is conducted away from the filament through the filament supports and leads	3 per cent
Heat dissipated by gas convection and conduction	8 per cent
Heat radiated by the bulb	8 per cent

Hence from a clear-bulb, gas-filled lamp about 90 per cent of the total energy is in radiant form; *i.e.*, all of it except that dissipated by the filament supports and leads, and by the gas. In burning a bare lamp in an interior its radiant energy will not be effective in raising the temperature in that interior until it has been intercepted by an absorbing surface which in turn will dissipate the heat by convection. The radiant energy will, however, increase the feeling of warmth to the human body by its radiant effect. Practically all surfaces such as building materials, room furnishings, and lighting equipment ab-

[1]The Tungsten Lamp, by W. E. Forsythe and E. M. Watson. (*Journal of Franklin Institute* Vol. 213, No. 6, June, 1932, p. 623).

sorb a part of the wave energy reaching them from a burning lamp. As far as the lighting equipment is concerned the amount of radiant energy intercepted by it will depend upon its type.

One of the most generally used types of lighting equipment is the white glass enclosing globe. If a lamp is placed in one of these units it is estimated that on the average the globe will absorb something of the order of 15 to 25 per cent of the 81 per cent radiant energy reaching it from the lamp filament and nearly all of the 8 per cent reaching it from the bulb. Hence, when the lamp bulb is placed in a white glass enclosing globe there will be 60 to 65 per cent of the energy dissipated in radiant form. The remaining 35 to 40 per cent of the energy is dissipated from the lamp and enclosing globe by conduction and convection. Theoretically, it is only this heat that can be removed from the lighting units by a circulating air system and some attention has been given to this possibility in order to reduce the heating effect in a room.

If no attempt is made to intercept and carry away any of the heat generated by the lamp, the question arises as to what portion of it is actually effective in raising the room temperature or in augmenting the load on the cooling system in an artificially cooled building. At first thought it would appear that the heat equivalent of the entire energy input to the lamps, amounting to 3.415 Btu per watt per hour, must inevitably be regarded as being effective. In fact, that assumption is ordinarily made in engineering calculations for a cooling system. However, a close examination reveals that a fairly good safety factor is being applied.

It is to be remembered that the engineer is concerned primarily with heat as it affects human comfort, and this may be either in the form of radiant heat or as heat in the surrounding air. Radiant heat which comes from a lighting unit or is re-radiated from the walls or ceiling is instantaneously effective in producing a feeling of warmth. Its effect can be counteracted by maintaining a lower air temperature in a building which is artificially cooled. However, not all of the radiant heat from the lamps becomes a load upon the cooling system. A portion of it passes through the windows to the out-of-doors and some of that which strikes the walls, ceiling, and floor is absorbed by the building structure and partially dissipated during the cooler night hours.

The ceiling height of the room has an important bearing upon this problem. The direct radiant heat from the lights will be much more noticeable in a low-ceiled room because its effect varies inversely as the square of the distance of the human occupants from the heat source. Another effect of the ceiling height is due to stratification of the air. The heat from the lamps which is intercepted by the glassware surrounding them, amounting to 35 to 40 per cent of the total energy of the lamps, causes the air to be heated surrounding the lamp and forms a stratum of very warm air near the ceiling. If this stratification is not disturbed by forced air currents it will not affect the comfort of the occupants below, especially if the room has a high ceiling. If, however, the cooling system is designed to introduce the air at high velocity several feet above the floor, the stratification is disturbed and the lighting fixtures are swept by the currents of incoming cool air and their heat, absorbed by the incoming air, becomes a part of the load on the cooling system.

Room Temperature Rises

In order to investigate the relation between the temperature rise and the wattage capacity of the lighting system in a typical commercial office a space was selected in an office building where the lighting capacity could be varied. This particular space had approximately 300 sq ft (17 ft by 17½ ft) of floor area with an 11-ft 9-in. ceiling, and had two outside windows, each of which was approximately 3 ft wide and 6½ ft high. The results of the investigation are shown in Fig. 4, Curves A and B. Curve A represents the temperature rise for various wattage capacities of lighting when the room is closed as much as possible by keeping the windows and doors shut. Curve B represents the temperature rise on typical summer days when the office is ventilated by raising the lower sash of the windows and opening the transom over the door. For these tests the thermometers were suspended in a vertical position near the center of the room with the bulbs 3 ft above the floor. Temperature readings were taken for a period of 7 hours when a condition of equilibrium was being approached; that is the heat dissipated from the room was nearly equal to the heat input and there was consequently but little further rise to be expected.

From the curves it will be observed that in the typical closed room with no artificial or natural ventilation the temperature rise during the 7-hour period due to the lighting system is approximately ¾ F for each watt per square foot of electrical energy consumed (Curve A). On the other hand, with the windows and transom open for natural ventilation on typical summer days the temperature rise is of the order of 0.22 F for each watt per square foot of lighting capacity (Curve B). The lighting equipment consisted of four open-top, semi-indirect units at the ceiling supplemented by indirect floor standards to provide the higher capacities.

Temperature readings in the test room were compared

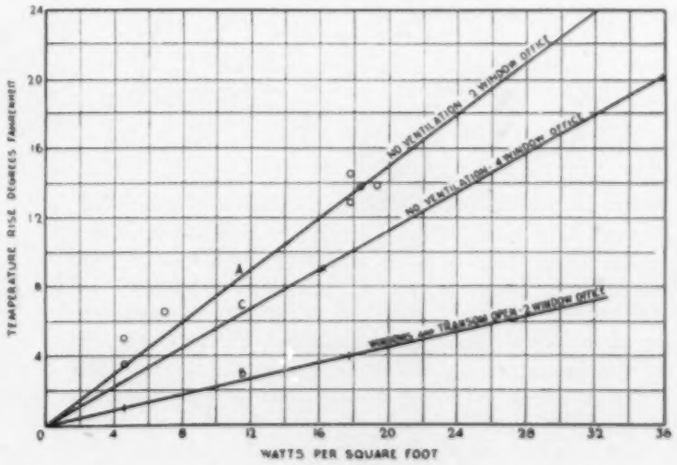


Fig. 4—Relation between the temperature rise during a 7-hour period and the wattage capacity employed for lighting in two typical office spaces

with those observed in an adjacent control room where no artificial lighting was used. The control room temperatures always remained practically constant during the test period. The rise in temperature in the test room was taken as the difference between its initial and final readings, and then this difference was corrected with any change which may have taken place in the control room.

The curves in Fig. 5 illustrate how the temperature in the test room rose with respect to the elapsed time during the 7-hour period when 7 w per square foot were used (Curve A) and when 19.4 w per square foot were used (Curve B. From these typical curves it is noted that approximately half the total rise occurred during the first hour.

The curves in Fig. 6 indicate that the outdoor temperature had little effect on the test room temperature rise during the 7-hour period when the windows were closed. It should, however, be mentioned that on the colder test days both the control room and the test room received heat from adjacent heated areas in the building so that comfortable basic temperatures were maintained. The test room curves represent conditions where 18 to 19 w per square foot

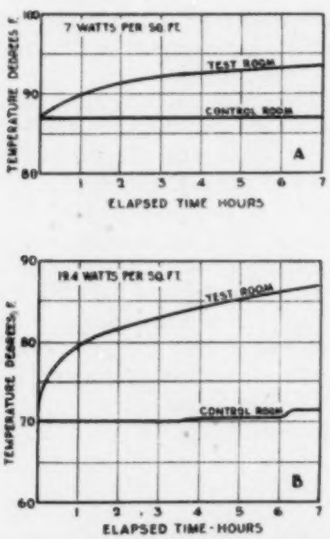


Fig. 5—Characteristic curves indicating that approximately half the total temperature rise for the 7-hour period occurred during the first hour

were employed (approximately 75 footcandles of illumination) with outdoor temperatures ranging from 32 to 87 F. Temperature rises of the order of 13 to 14 F were obtained.

On the other hand, with 18 to 19 w per square foot for lighting the room when the windows were open on a typical summer day the rise in temperature was of the order of 4 to 5 F for the 7-hour period. This is shown by Curve B, Fig. 4.

For general lighting throughout interiors such as stores, offices, and factories the electrical capacity in actual use at the present time is 1 to 2 w per square foot of area. With 2 w per square foot it is observed from

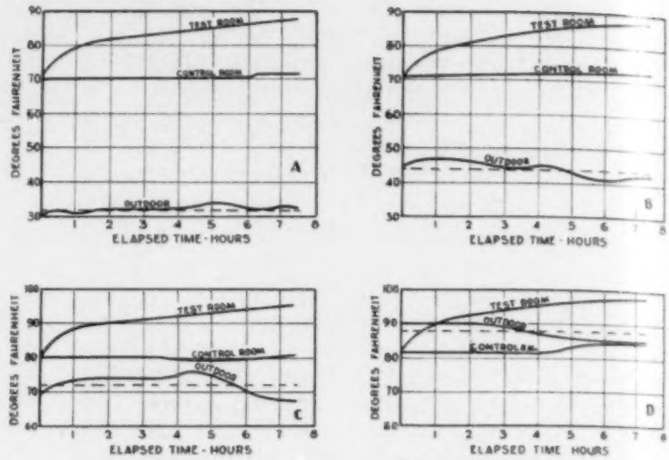


Fig. 6—Curves A, B, C and D show temperature rises when 18 to 19 w per square foot capacity was used for lighting and with outdoor temperatures of 32, 44.5, 72 and 87.5 F respectively

Fig. 4 that the temperature rise in the closed room is 1½ F while in the ventilated room during the summer time it is only about ½ F. These temperature rises are so small that for uncooled buildings little notice has been given to the heat generated by the artificial lighting system.

A few typical examples of the relative importance of the several items which constitute the cooling load in different kinds of buildings are shown in Table 2. These are values which were calculated in the process of designing the cooling systems. The first building in the table is a sales office of an electrical company. The show windows are not partitioned from the salesroom and the heat from the high level window lighting must therefore become a load on the cooling system. The building is shaded by adjacent buildings so that the sun never shines through the windows and the external load is therefore small. This combination of conditions causes the lighting to be 50 per cent of the total calculated load.

At the other extreme is the theater, in which the light-

Table 2—Typical Examples Showing the Relative Importance of Factors Affecting the Cooling Load in Different Classes of Interiors

Type of Interior	Lighting, Watts per Sq Ft of Floor Area		Relative Factors in Per Cent				
	Installed	Assumed in Use	Lighting[a]	Solar Radiation Through Windows	Heat Conducted Through Walls and Windows	Infiltration of Outside Air	Occupants[b]
Sales office of electrical company with show window lighting	6.9	4.4	50	0	2	46	2
Office building							
With awnings	7.0	3.1	37	0	11	37	15
Without awnings	7.0	3.1	26	33	4	26	11
Office building with glass block walls	4.5	1.8	31.4	22.7[c]	4.6	28.2	13.1
Cafeteria	0.9	0.9	9	0	6	28	54
Theater		2.6	5	0	5	45	45
Research Laboratory	2.0	0.7	8.6				

[a] Includes fan power load.
[b] Includes latent heat load.
[c] Includes heat gain by conduction through the glass blocks.

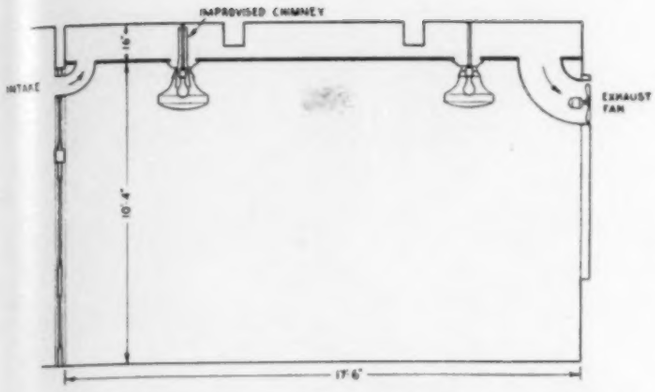


Fig. 7—Cross-sectional view of typical office used in test showing false ceiling with openings above the lighting units and location of intake and exhaust for ventilating air

ing is almost negligible as a source of cooling load. The load imposed by the occupants and by the introduction of outside air constitutes most of the total load. In the case of one of the office buildings, the use of awnings eliminates the solar radiation through the windows as a source of load to the cooling system, thus increasing the relative effects of the lighting and other factors.

The values in Table 2 represent the maximum cooling load, which is used in selecting the size of the apparatus. It is of course true that, on the cooler days in summer, the load caused by the lighting, though no larger in absolute quantity, is a greater proportion of the total. The figures referring to the lamps are based on the assumption ordinarily made that the entire heat equivalent of the energy input to the lamps which are assumed to be actually in use becomes part of the cooling load. Therefore these values show the maximum effect of the lighting on the cooling system. It may then be concluded that the lighting is responsible for a part of the cooling load varying between 5 and 50 per cent of the total, depending upon the type of building. Recognizing this fact, thought has been given to ways of eliminating the heating effect from lighting units in some manner such as having the heat diverted directly from the source into a ventilating duct where it will be dispensed with or utilized as may be desired.

Diverting Heat and Absorbing Radiant Energy

In order to take advantage of the natural tendency of heat to rise into higher altitudes, an experimental lighting system was installed which employed four 20-in. enclosing glass globes with hollow fitters connected to openings in a false ceiling. Fig. 7 illustrates diagrammatically the installation which was made in a typical office room. An exhaust fan was used to remove the air from the space above the false ceiling; the volume of air removed and the temperature rise between the intake and the exhaust air was observed. The results of the test are shown in Table 3, with a total of 4163 w actual consumption at the sockets and with the room doors and windows closed.

Circulating Air Through a False Ceiling

The data from Tests 1 and 2 indicate that about 30 per cent of the heat generated by the lighting units can be removed by permitting the heated air to rise above a false ceiling where it can be carried away by a ventilating system. Although a greater portion of this heat was removed directly from the lighting unit, yet a small portion of it was no doubt conducted from the room through the false ceiling into the ventilating air. It was felt that further experimental work should be done to obtain more accurate and reliable data on the maximum amount of heat which can be removed directly from a light source by circulating air through it, and therefore another series of tests were made, as will be described later.

By removing approximately 30 per cent of the heat the temperature in the room increased from 80 to 91.5 F, resulting in a rise of 11.5 F, during the 7-hour period. Under similar conditions but without the ventilating system operating above the ceiling, the temperature in the room increased from 69.5 F to 88.5 F, a net rise of 19 F during the 7-hour period. In this latter test it is believed that the 19 F rise is 4 to 5 F higher than it should have been. This opinion is based upon the fact that for this particular test the initial temperature was 69.5 F while the control room registered 75 F. The test should have been repeated with approximately the same starting temperature for the two rooms but the equipment was dismantled before the discrepancy was noted. In a majority of the other tests the difference between the starting temperatures of the two rooms was less than 1 F and in no case did it exceed 2 deg.

Table 3—Data and Results Obtained from Tests Conducted in Room Shown in Fig. 7

	Test No.					
				With Chimney Sleeves on Lamp		
	1	2	3	4	5	6
Change of Air, cfm.....	442	342	No change	442	342	No change
Intake Temp., F.....	80.3	79.4		78.5	82	
Exhaust Temp., F.....	89.5	90.7		85.5	89.4	
Ventilation Air Rise, F..	9.2	11.3		7	7.4	
Btu/ min Removed.....	72	68.5		54.7	44.8	
Btu/ min Generated...	237.3	237.3	237.3	237.3	237.3	237.3
Per Cent Btu Removed.	30.3	28.9		23.1	18.9	
Test Room Temp. Rise after 7 hours, F.....	11.4	11.7	19	10.5	11.7	15.6

In attempting to remove heat from the lighting units in the foregoing manner several tests were made with cylindrical and conical-shaped tubes (see Fig. 7) inside the enclosing globes over the lamps to create a greater flow of air upward around the lamp bulbs. The results of this investigation are given under tests Nos. 4, 5 and 6 in Table 3 which indicate that the flow of heat from the lamp inside the enclosing globe was actually retarded rather than assisted by the improvised chimneys. For example, in comparing the results of Test No. 4 with No. 1 it will be noted that only 23.1 per cent of the generated heat was removed when the chimney was used, while without it 30.3 per cent was removed. The test room temperature rise was found to be about the same in both cases. The opening into the ceiling was 4⅞ in. diameter and, when a thin-walled cylinder 3 in. in diameter with conical shaped lower section was placed through it and around the lamp bulb, the resistance to the air

flow was apparently too severe to be overbalanced by the theoretically greater draft action which should have been created by the chimney. The cross-sectional area of the chimney was made approximately one-half the total area of the opening so that the air flow would be as free as possible.

Considerable time and effort was spent to determine the most effective size as to diameter and length and the best shape of the chimneys as well as their proper location in the lighting units. On this phase of the investigation the effectiveness of the chimney was judged by observing the temperature at a point on the side of the enclosing globe. The point for observing the temperature was taken about half way down the side of the globe so that none of the various sizes and shapes of sleeves used intercepted the direct radiant energy between the light source and the thermometer. The bulb of the thermometer was placed in direct contact with the enclosing globe and held in the same position throughout the investigation. Then, for any given condition of ventilation in the room or above the ceiling, temperature readings were observed on two lighting units, one unit being the control unit and the other one the test unit with the sleeve in it. The readings indicated that different sizes and shapes of sleeves had very little effect on the temperature of the globe, assuming a minimum size of sleeve of 3-in. diameter and a sufficient length to extend from slightly below the socket to a point well above the false ceiling.

Tests No. 4 and 5 having forced ventilation above the ceiling show room temperatures about 5 F lower than Test No. 6 which represents a similar test but without forced ventilation.

Maximum Heat Removed by Forced Ventilation

As mentioned in the foregoing, it was questionable as to just how much of the approximately 30 per cent heat removed by Tests No. 1 and 2 was due to ventilating the lighting units and how much was due to conduction through the ceiling. Further tests were therefore made

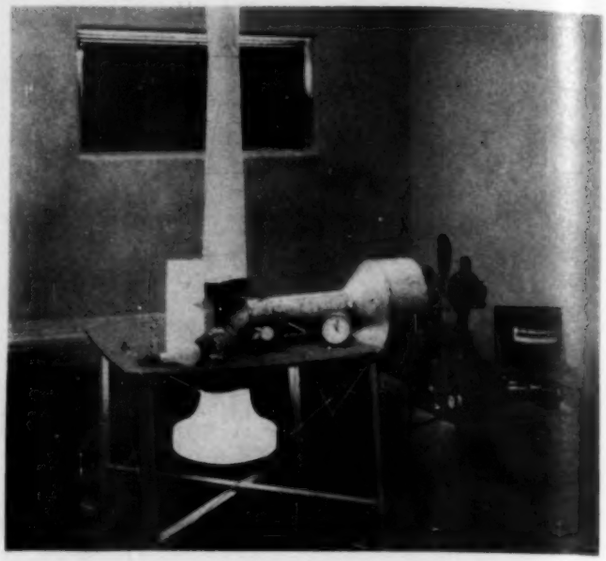


Fig. 8—Arrangement of equipment to determine the maximum amount of heat removable by forcing air through an enclosing globe when suspended in a room having a constant temperature

with equipment as shown in Fig. 8 to determine the maximum per cent which can be removed by circulating air through lighting units of a type and design in general use suspended below a ceiling. This equipment consisted of a 14-in. white glass enclosing globe having a hollow fitter through which intake and exhaust air ducts were brought to the lamp. For the investigation the enclosing globe was suspended in a room where the temperature was held constant while different volumes of air were passed through the globe in which different sizes of lamps were burned. The curves in Fig. 9 show how the ventilating air temperature increased as the volume of the ventilating air decreased for lamps of the 75, 150 and 300-w sizes. From the temperature rise and volume of the ventilating air the heat removed was computed. Fig. 9-B shows the heat removed in per cent

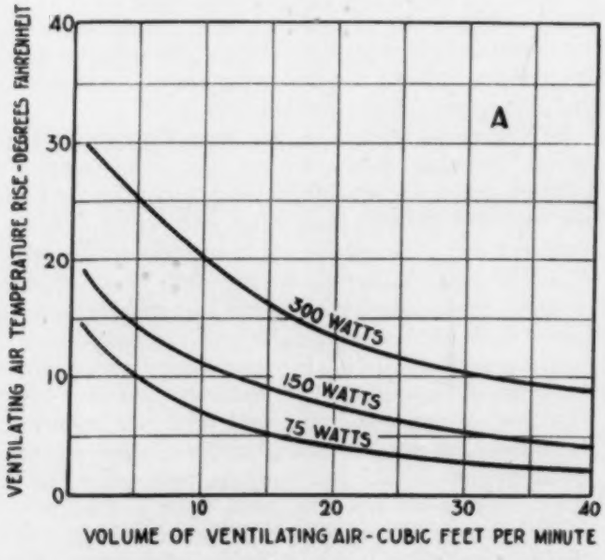


Fig. 9A—Relation between volume of ventilating air and its temperature rise when lamps of the 75, 150 and 300-w sizes are used in equipment shown in Fig. 8

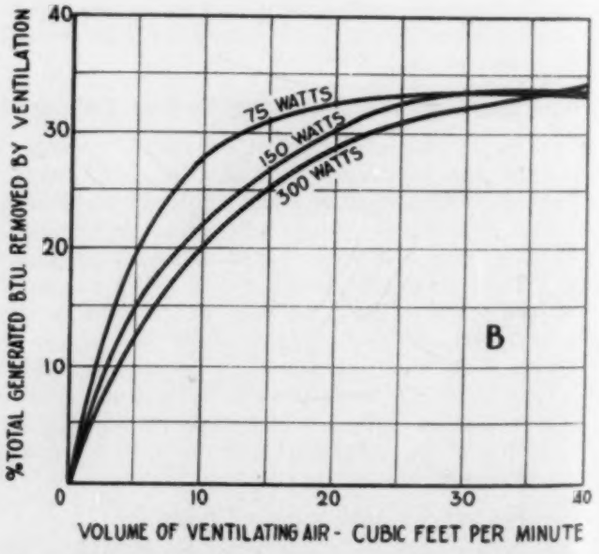


Fig. 9B—Relation between volume of ventilating air and per cent Btu removed as computed from data in Fig. 9A

of that generated by the lamp for different volumes of ventilating air. It will be noted that these three curves practically coincide when 35 to 40 cfm of forced ventilation are used. Under this condition it appears impossible to remove by a current of air more than about one-third the total heat generated by the lamps. This amount checks closely with the theoretical percentage mentioned earlier in this paper.

Another test was conducted with the equipment shown in Fig. 8 by placing the 14-in. enclosing globe in a cubical box 34 in. on a side as illustrated in Fig. 10. The cubical box was made of ½-in. thick insulating board which is a comparatively poor conductor of heat. The object of this test was to compare the air temperatures in the box when the lamp was ventilated, with those without ventilation. Lamps of the 40, 75, 150 and 300-w sizes were used with approximately 75 cfm of ventilating air passing through the enclosing globe. In Fig. 11 the solid-line curves represent the temperature rise during a 4-hour period when the lamp was ventilated and the dotted-line curves represent the temperature rise during a similar period with no ventilation. A 4-hour period of operation was chosen as it was found that it required about this length of time for the temperature in the box to approach a maximum. The data in Table 4 based on temperature readings at the end of the 4-hour period indicate that the box temperature was approximately 40 per cent lower when the lamp was ventilated than it was with no ventilation. If this box could be considered as a small room having ½-in. thick insulating board as side walls, floor and ceiling, and no openings (and it is assumed that all the radiant energy must be absorbed somewhere within the room), then it is found that a maximum of approximately 40 per cent reduction in temperature can be obtained by circulating cooled air around the light source.

Further data from the box tests permitted the determination of the heat removed from the lighting unit by the air duct. When approximately 75 cfm were used in the air duct, the temperature rises of the air at the end of the 4-hour period are as given in Table 4. From these data the heat removed from the lamp was computed

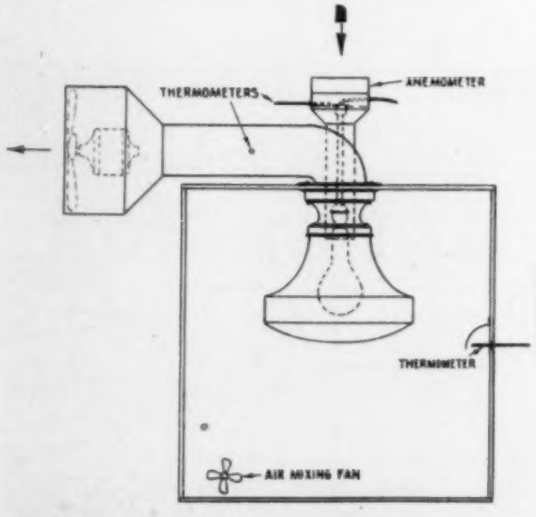


Fig. 10—Cross-sectional view of insulating board box and equipment employed for comparing the temperature rise in the box when the unit is ventilated and non-ventilated

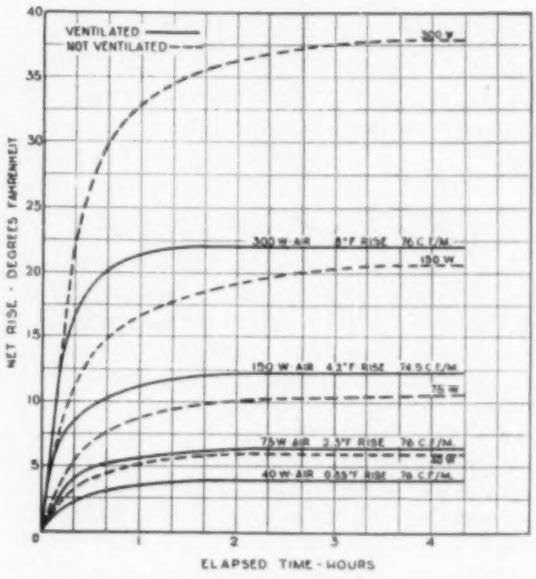

VENTILATED
NOT VENTILATED
300 W
300 W AIR 8°F RISE 76 C F/M
150 W
150 W AIR 4.2°F RISE 74.5 C F/M
75 W
75 W AIR 2.3°F RISE 76 C F/M
40 W
40 W AIR 0.85°F RISE 76 C F/M
NET RISE - DEGREES FAHRENHEIT
ELAPSED TIME - HOURS

Fig. 11—Curves showing net temperature rises during a 4-hour period in test box with four different sizes of lamps in lighting unit when ventilated and not ventilated

Table 4—Data and Results Obtained from Tests with a 14-In. White Glass Enclosing Globe in a 34-In. Cubical Box

Lamp Size Watts	Non-Ventilated Lighting Unit Net Temp. Rise in Box, F	Ventilated Lighting Unit (Approx. 75 cfm)			Btu per Hour Generated by Lamp	Percentages	
		Net Temp. Rise In Box, F	Temp. Rise in Air Duct, F	Btu per Hour Removed by Air Duct		Temp. Rise in Box Ventilated to Non-Vent.	Heat Removed to Heat Generated
300	37.9	22	8	608	1024.5	58.0	59.4
150	20.7	12.3	4.2	326	512.3	59.4	63.5
75	10.5	6.3	2.3	182	256.1	60.0	71.0
40	6.2	4.0	0.85	67	136.6	64.5	49.3
Average						60.5	60.8

and found to represent approximately 60 per cent of the total generated. This percentage was considerably higher than that found in the previous test for the unit (Fig. 8) when operated in a constant temperature room. This difference was probably due to the abnormal conditions of having the lighting unit confined within the box where the temperature of the enclosing globe was obviously much higher.

Heat Resisting and Absorbing Media

It is generally known that water will readily absorb practically all the long-wave infra-red radiation, that is, wave lengths longer than 1.4 to 1.6 μ (microns). On the other hand, water transmits a high percentage of the energy appearing in wavelengths shorter than 1.0 to 1.2 μ which includes the visible energy. Fig. 12 shows the relative energy at different wave-lengths as generated by a 500-w gas-filled incandescent lamp. Superimposed on this curve is a transmission curve for energy passing through 1 cm (centimeter) of distilled water. From this it is obvious that lighting units with cold water jackets instead of cold air would greatly facilitate the elimination of heat immediately at the light source. A light source adopting this feature was developed several years ago[2]

[2]Water Cooling of Incandescent Lamps, by N. F. Gordon (*Society of Motion Picture Engineers Transactions*, Vol. 14, No. 3, p. 332).

and recent designs based on the original features have found practical application for the lens-type photographic enlargers.

The unit consists of a distilled water jacket surrounding the lamp and a submerged coil through which cold water is continuously circulated. Tests on such a device employing a 1500-w projection type lamp indicate that 75 per cent of the radiant energy is absorbed by the water while the sacrifice in total light output is only of the order of 20 per cent; although this method of dispersing the heat from an artificial lighting system is entirely practical under certain conditions, yet its general application for large interiors appears somewhat remote at the present time.

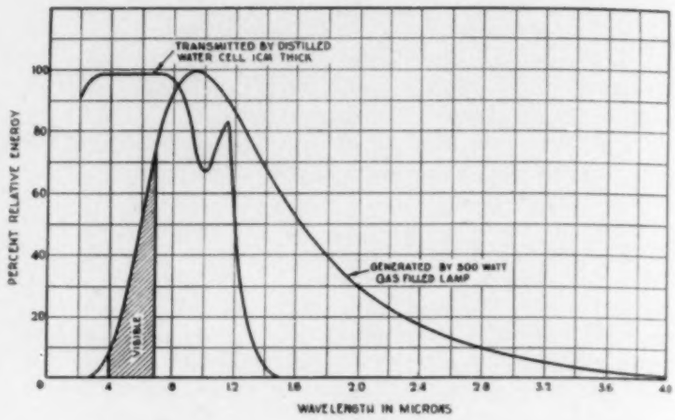

TRANSMITTED BY DISTILLED WATER CELL 1CM THICK
GENERATED BY 500 WATT GAS FILLED LAMP
VISIBLE
PERCENT RELATIVE ENERGY
WAVELENGTH IN MICRONS

Fig. 12—Relative energy at different wave-lengths from a 500-w gas-filled incandescent lamp and the per cent of this energy transmitted through 1 cm of distilled water

Another medium for absorbing radiant energy from incandescent lamps has been developed[3] in a heat-resisting, heat-absorbing glass having a 2-millimeter thickness which will absorb 79 per cent of the total energy generated from a 500-w projection lamp. The light transmission of this special glass is still maintained fairly high, being approximately 75 per cent. Plates of this glass are available for use in false ceilings above which artificial lighting systems can be installed. With such a system a circulation of air could be produced to cool the glass plates and thereby to provide a method of eliminating a high percentage of the radiant energy from a lighting system before that energy is set free in an interior.

Practical Applications

Many lighting systems have in the past been installed with ventilation ducts to eliminate the heat. Such systems have always been associated with high levels of illumination, 50 to 100 footcandles or more, as generally used in display windows, hospital operating rooms and photographic studios. An example of the removal of lamp heat by air circulation is illustrated in Fig. 13. This shows a sales office of an electrical company having high level window lighting in which the windows are not partitioned off from the sales space as is usually the case. The heat generated by the lamps would therefore be an expensive factor in the cost of operating the cooling system installation.

The lighting units are enclosed in metal boxes, each of which has a duct which supplies cool air drawn from the basement. The warm air is discharged to the out-of-doors during the summer when the artificial cooling system is in use. In winter, by a change of dampers, the heat is diverted to the second floor of the building where it is utilized for part of the heating requirements. Means are provided for installing a circulating fan but the gravity circulation alone appears to be sufficient. This installation was made partly for economic reasons in view of the cost of summer cooling, and partly to permit better control of temperatures during the heating season.

A recent article[4] gave details of a ventilated lighting system in a Philadelphia display room having dimensions of 40x110 where 40 footcandle illumination was supplied from eight large alabaster diffusing glass units each containing thirteen 500-w incandescent lamps and four 450-w mercury lamps, making a total of 8.3 kw. With a heat dissipation of 15.5 w per square foot of floor space, the discomfort resulting therefrom had to be prevented by the air-conditioning equipment, which would have entailed with this lighting load an additional investment and operating expense that was unwarranted. Consequently a partly closed cylinder of heat-absorbing glass was placed around each incandescent lamp, through which air was drawn by a ventilating duct system connected to each fixture. It was found necessary to operate the ventilating fan for at least 5 min after the lights were turned off to avoid breakage of the alabaster diffusing glass by the sudden storage of heat, which would otherwise have occurred. The ventilator withdrew the air from each fixture at the rate of 600 cfm, discharging it outdoors in warm weather. When heat was needed in the building the air withdrawn from the fixtures was delivered to the air-conditioning system in such proportions as required. This arrangement reduced the cooling cost of the air-conditioning system one-half

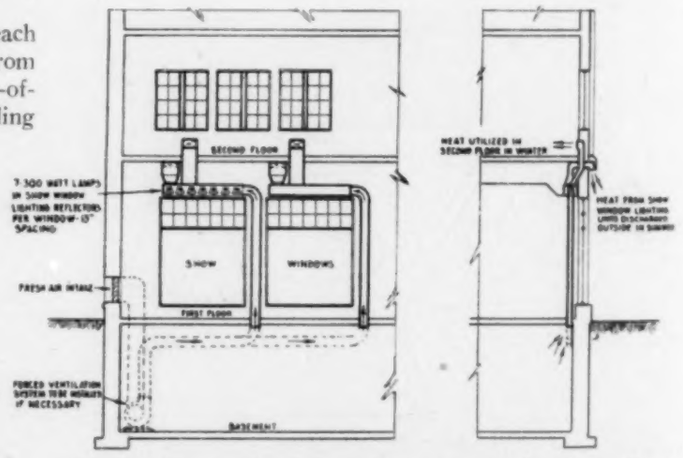


Fig. 13—Sketch showing ventilating system for show-window lighting; dispensing with heat in summer and utilizing it in winter

[3]Color Glass Filters (Corning Glass Works Bulletin No. C-112).
[4]Removing Heat from High Intensity Lighting (*Electrical World*, April 15, 1933, p. 480).

and the steam-heating system was only used on the very coldest days.

Another practical solution of eliminating heat was described[5] in connection with a clothing store located in Atlanta, Ga., which was already air-cooled. The store was relighted by increasing the lighting load on the first floor from 4000 to 22,000 w. Due to the fact that the first floor of the store was air conditioned, the problem of increasing the lighting load from 4 kw to 22 kw without affecting the capacity of the air-conditioning equipment was overcome by building the fixtures (two luminous projecting beam type) the entire length of the area to be lighted and by placing a filter on one end of the fixture and a positive pressure fan on the other end. The amount of additional heat was eliminated by this design. The fan was separately controlled so that in the mild winter months it could be cut off, and only on a few occasions has the heat from the lighting fixture been insufficient to heat the store. Care should be taken in the construction of the fixture to make it fairly tight so that it will act as a duct and extreme care must be taken in selecting the size of the ventilating fan.

In any room where warm air is displaced by cool air, it would seem logical to those who are inexperienced in the technique of air conditioning that the cool air should enter the room near the floor and the warm air be withdrawn at the ceiling. Quite the reverse is the modern cooling system which is designed to have the incoming air enter the room near the ceiling. This, of course, has certain advantages, the most important of which is the elimination of drafts and uncomfortably cool conditions for the occupants of the room.

The method of reheating the cold air to a comfortable temperature before it is released is a subject of much controversy. In general, the reheating is accomplished by mixing the dehumidified cold air with warm air which has either been by-passed around the cooling coils or which has been obtained from outdoors. In either case the volume of warm air must be sufficient to properly temper the air before it reaches the occupants of the room. In many cases at least a part of this tempering is accomplished by releasing the dehumidified cold air near the ceiling and directing it around the lighting units where an adequate amount of heat is absorbed. Estimations indicate that all the sensible heat generated by a lighting system could be used for this purpose unless it exceeds about one-third the total sensible heat gain for the interior.

New Light Sources

The curves in Fig. 1 show the gradual increases in efficiency for incandescent lamps which have been made during the past two decades. Although it is expected that further advances will be made, yet it is recognized that there is a definite limit to them because the efficiency of a lamp depends on the operating temperature of the tungsten filament, which in turn is limited in order to keep a safe distance below its melting point.

A new outlook with comparatively unlimited possibilities has recently appeared in the form of electric discharge lamps employing mercury vapor. These Type H lamps have entirely different operating characteristics and color quality of light from incandescent lamps, but they are suitable for a wide range of industrial and commercial lighting applications. They are now available in only two sizes, the 400-w (Type H-1) and the 250-w (Type H-2). Of outstanding importance is the fact that the 400-w lamp has an efficiency of 40 lumens per watt, and the 250-w lamp has an efficiency of 30 lumens per watt. These light output efficiencies can be compared with the 19 to 20 lumens per watt obtained from incandescent sources of an equal wattage. Mercury lamps (Type H) will therefore provide illumination with a heating effect which would at first appear to be considerably less than that required from an incandescent system. However, their color when used alone is not generally acceptable for stores and offices. Hence, for these interiors, mercury lamps are usually combined with incandescent lamps so that the combination gives substantially a daylight effect. Where this is done there is usually no saving in wattage.

The color of the light emitted by mercury lamps has the characteristic mercury vapor line spectrum in contrast to the continuous spectrum of the incandescent lamp. This means that all the light from the mercury arc is represented by only a few bands which produce yellow, green, and blue light. Red is practically absent. The result is that although the light itself has a bluish-white cast, materials of certain colors such as the reds are not recognized, while yellows and greens are emphasized. However, in the work spaces where whites, greys and other neutral tones predominate, Type H mercury lamps are satisfactory.

The operating characteristics of mercury lamps are such that each lamp requires a special transformer or regulating device for starting and operating. Also these lamps require several minutes to attain full brilliance after being turned on.

Another type of lamp now under development which promises to find application in auditoriums, theaters, restaurants, etc., where lumiline incandescent lamps are now being used, is the fluorescent lamp, which is in effect a mercury lamp similar to the Cooper-Hewitt but smaller. The lamp itself is not very efficient, but inside the tubular bulb there is a great deal of shortwave, ultraviolet energy which does not appear in the form of light because it is generated below the visible spectrum. Not long ago it was discovered that when a powder which fluoresces under ultraviolet radiation is placed inside a mercury lamp, the powder turns the short wave energy into visible energy which produces a considerable amount of light. Since this discovery, there has been a search for powders that would produce white light efficiently and also other colors, for decorative purposes. Present developments indicate extraordinary increases in efficiency over that obtained from the present lumiline incandescent types. Fluorescent lamps are not commercially available as yet.

Radiant Energy From Mercury Lamps

In analyzing the distribution of energy dissipated from 300- to 500-w incandescent lamps, it was found that approximately 90 per cent of it occurred in the form of radiant energy. In comparison to this about 70 per cent is found for the 250-w Type H lamp and 65 per cent for

[5] Atlanta Store Acknowledges Added Business Through Improved Lighting (*Magazine of Light*, May, 1937, p. 20).

the 400-w Type H. The fluorescent lamp (15-w 18-in. white experimental) dissipates about 50 per cent of its heat equivalent in the form of radiant energy. Using these percentages the heat dissipated in radiant form can readily be computed for each light source (see Table 5). The last column in Table 5 shows the comparative radiant energy in watts when adjusted to a 10,000 lumen light output basis.

The radiant energy would, of course, be reduced somewhat if the lamps were placed in lighting units which would intercept and re-radiate it. Although no definite data are available, it is estimated, for example, that when a 400-w mercury lamp is placed in a lighting unit less than half its total energy will be dissipated in radiant form. In other words, a comparatively high percentage of the energy would be conducted and convected away from a mercury lamp in a lighting unit. Mercury lamps of the future may not only have an advantage over incandescent lamps from an efficiency point of view, but also from the cooling and ventilating viewpoint, because a greater proportion of their heat energy may be carried out of the building by air drafts around them when placed, for example, above skylights or behind built-in glass plates. In some cases, however, air drafts may affect the operating efficiency of mercury lamps, but experience data along this line are lacking.

Conclusions

For lighting systems employing the usual type of luminaires placed below the ceiling, it is mechanically possible to install a ventilating system with air ducts to the units so that a maximum of about one-third the total generated energy can be removed. With the so-called *built-in* types of lighting equipment where a free circulation of air is provided around the lamp, the amount of heat carried off will depend upon the heat transmission characteristic of the glass through which the light enters the room. If this glass is of the heat-resisting heat-absorbing type and is itself cooled, then at least twice as much can be removed by the ventilating air as found possible from the usual types of lighting units suspended below the ceiling.

If none of the energy generated by the lighting is carried off by a ventilating system it is estimated that from 50 to 90 per cent of it will eventually become a part of the cooling load. Inasmuch as a maximum of only one-third the total energy can be removed from the usual types of lighting equipment, it is the opinion of the author that the cost of an elaborate auxiliary ventilating system for this purpose would not in general be economically justified for the present standards of lighting whose requirements, as indicated in Table 1, seldom exceed 5 w per square foot of floor area. In other words, for these general interiors it would be less costly to provide sufficient cooling capacity to offset the heat generated by the lamps as well as that from other sources.

However, for interiors where high levels of illumination are used which require 10, 15 or more watts per square foot of electrical capacity, careful consideration should be given to a separate ventilating system which not only carries off the readily removable one-third of the heat energy, but also makes use of heat-resisting glass to intercept a considerable portion of the radiant energy.

Table 5—Approximate Radiant Energy from Incandescent and Mercury Type H Lamps[a]

Light Source	Light Output (Lumens)	Radiant Energy (Per Cent)	Watts Dissipated in Radiant Energy	Radiant Energy (Watts) for 10,000 Lumen Light Output
500-w incandescent lamp	10,000	90	450	450
400-w mercury	16,000	65	260	163
250-w mercury	7,500	70	175	233
15-w fluorescent (white experimental)	450[b]	50	7.5	166

[a]Evaluated in watts dissipated for equal light production of 10,000 lumens.
[b]Assumed Output.

In the foregoing discussion the problem of removing heat generated by the lighting system is of importance in northern latitudes for only 3 to 4 months each year. During the other 8 to 9 months this heat may be of considerable value in contributing to the heating system for the building.

Acknowledgment

The author wishes to acknowledge his appreciation of the valuable suggestions received from several heating and ventilating consultants and particularly his indebtedness to J. H. Walker, of the Detroit Edison Co., for data and assistance and to the *Illuminating Engineering Society* for data obtained from their *Transactions*.

Open House at A. and M. College of Texas

On December 15 an Open House was held at the A. & M. College of Texas in order to acquaint those interested in the cotton industry with the purposes of the cotton fiber and spinning research conducted at the College.

The cotton fiber and spinning research work being done at the College is a cooperative project between the Bureaus of Agricultural Economics and Plant Industry of the U. S. Department of Agriculture, and the Agricultural and Engineering Experiment Stations of the A. & M. College of Texas. An annual appropriation secured from Congress for cotton fiber and spinning research was divided between the laboratories at Clemson College, South Carolina, and Texas A. & M. College. The first year's appropriation for the Texas Station was used largely for the purchase of a complete line of yarn manufacturing machinery, yarn testing equipment and an air conditioning system for the spinning laboratory. The spinning and testing laboratories are now completely established and are being operated by a staff of eleven persons.

The first spinning tests undertaken were designed to form a part of the Regional Variety Studies now in progress by the Bureau of Plant Industry and the Agricultural Experiment Stations of the various cotton growing states. Work was begun with the crop of 1935, and spinning tests have been completed on seven replicate samples grown at each of eight stations scattered over the cotton belt.

Pittsburgh Experiment Station of the U. S. Bureau of Mines where the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS is located

Draft Temperatures and Velocities in Relation to Skin Temperature and Feeling of Warmth

By F. C. Houghten* (*MEMBER*), Carl Gutberlet** and Edward Witkowski*** (*NON-MEMBERS*)
Pittsburgh, Pa.

DRAFTS are probably the source of more complaints directed against ventilating and air conditioning systems than any other defects. Notwithstanding this fact, the engineer has no way of evaluating what constitutes a draft other than his own personal feelings. There is even a lack of understanding of just what is meant by a draft. Literally the term means a movement of air. Probably most complaints of draft result from excessive movement of air.

The sensation commonly called a draft is a local sense of coolness affecting some part of the body, compared with the general feeling of warmth of the body as a whole. It is doubtful if a person can distinguish between a local sense of coolness caused by air motion and a local sense of coolness resulting from contact with air having a lower temperature. A person may also experience a local sense of coolness due to radiation from a portion of his body to a cooler surface, such as a cold wall or window. This feeling may also be interpreted as a draft.

From these considerations, the term *draft* as applied to conditions of the atmosphere within an occupied space may be accepted as meaning any local sense of cooling of a portion of the body,—caused either by an excessive movement of air of normal temperature, by air having a normal velocity but a lower temperature, by excessive radiation to a cold surface, or any combination of these three effects. The problem of determining what combinations of temperature and movement of air constitute drafts was recently assigned to the Research Laboratory by the A.S.H.V.E. Technical Advisory Committee OH-22 on Comfort Air Conditioning, consisting of the following: C. Tasker, *Chairman*, A. E. Beals, F. R. Bichowsky, Thomas Chester, F. E. Giesecke, Elliott Harrington R. E. Keyes, A. B. Newton and C. P. Yaglou.

The study carried out during the past year has been directed toward the development of an understanding of the effect of low temperature and velocities of air as they determined a sense of coolness or a draft on the ankle and the back of the neck. In order to carry out the study, a person seated in a room of uniform temperature and humidity and normal air velocities ranging from 15 to 25 fpm, as measured by the Kata-thermometer

*Director, A.S.H.V.E. Research Laboratory.
**Research Asst., A.S.H.V.E. Research Laboratory.
***University of Pittsburgh Senior and Student Asst., A.S.H.V.E. Research Laboratory.

Presented at the 44th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., January, 1938.

had air streams of various temperatures and velocities directed against either the back of the neck or the ankle. Observations were made of the temperature of the skin of the affected part and the sensation of coolness or comfort observed by the subject.

Figs. 1, 2 and 3 show the test arrangements for making these observations. Air from a source of which the temperature and moisture content was independently controlled was supplied through a duct to the psychrometric chamber in which the subject was located. A Kata-thermometer located near the outlet, or nozzle, of the duct indicated the velocity of the air at this point and a thermocouple located near the same point indicated the air temperature. In carrying out studies of the neck the air velocity was adjusted to the desired value, indicated by the Kata-thermometer, without the subject being present and the orifice flow meter reading was noted. This nozzle air delivery was kept constant throughout the test. Also the temperature of the air at the outlet of the nozzle was maintained constant.

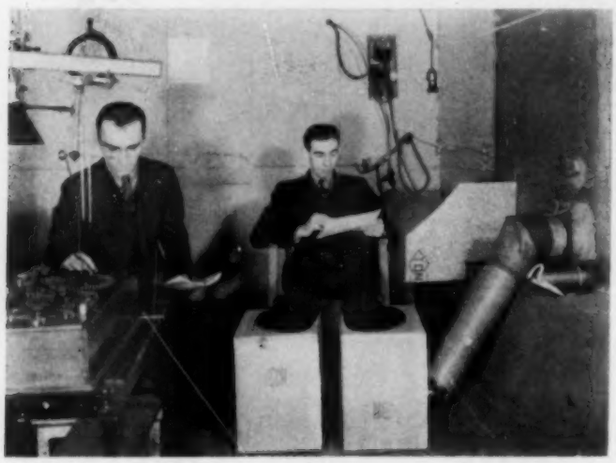


Fig. 1—Set-up for determining ankle temperature and feeling of draft

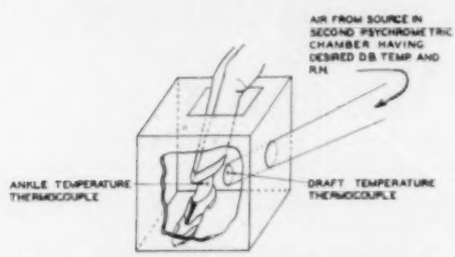

AIR FROM SOURCE IN SECOND PSYCHROMETRIC CHAMBER HAVING DESIRED D.B. TEMP AND R.H.
ANKLE TEMPERATURE THERMOCOUPLE
DRAFT TEMPERATURE THERMOCOUPLE

Fig. 2—Test arrangement for determining the effects of drafts on the ankle

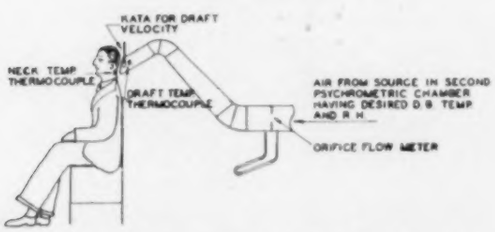

KATA FOR DRAFT VELOCITY
NECK TEMP. THERMOCOUPLE
DRAFT TEMP. THERMOCOUPLE
AIR FROM SOURCE IN SECOND PSYCHROMETRIC CHAMBER HAVING DESIRED D.B. TEMP AND R.H.
ORIFICE FLOW METER

Fig. 3—Test arrangement for determining the effect of drafts on the neck

After the skin temperature of the neck had been observed over a period of from 10 to 20 min, with little or no air flowing through the nozzle and with 70 F indicated by the thermocouple, the desired temperature and velocity were applied. The drop in skin temperature and the subject's expressed feeling of warmth of the affected part and of his entire body were recorded. In the study of the ankle each foot was kept in a corrugated paper packing box. Prior to the test, air was directed through the box so that the desired velocity was indicated by a Kata-thermometer, located where the ankle was later to be placed, and the flow meter reading was noted. After the temperature of each ankle had been observed for 10 to 20 min, or until equilibrium was established in air of practically zero velocity and 70 F temperature, air of the desired temperature and velocity was directed against one ankle while the other ankle remained in the controlled condition. The drop in skin temperature of the affected ankle below that of the ankle in the controlled condition, during a 30 min exposure, was taken as a measure of the cooling effect of the draft.

This paper reports the results of 141 determinations of the cooling effect of drafts on the neck and ankle. Since any method of making skin temperature observations must necessarily have an important bearing on the interpretation of the results, a study of different methods is discussed. This investigation, and others previously made at the Research Laboratory, afforded an opportunity to make many observations of normal skin temperatures of several parts of the body which should be of general interest. The results of these observations are also reported in this paper.

Measurement of Skin Temperatures

The measurement of the temperature of any surface is a difficult problem toward which a great deal of speculation and experimentation has been directed. All of the attendant difficulties and additional ones are met with in any attempt to determine the true temperature of the skin of the human body.

It is well to keep in mind that all that is done when measuring the temperature of any surface is to observe the temperature of the sensitive elements of a thermometer and pyrometer. Only through knowledge gained by experience can the sensitive elements of the instrument be so placed that it may be assumed to take on the temperature of the surface to be measured. In the case of observing the temperature of any surface, care must be taken, therefore, to place the sensitive thermocouple junction in contact with the surface so that, first, it will not interfere with the normal heat exchange between the surface and the air nor change the temperature of the surface; and, second, so that the instrument will take on the same temperature as the surface. Since both the surface to be measured and the couple junction are in contact with the air, the junction will be jointly affected, under any circumstances, by the temperature of the surface and the air. The closeness and area of the contact between the junction of the thermocouple and the surface to be measured is, therefore, important. When the surface, of which the temperature is to be measured is that of the human body, two additional factors affect the results: first, voluntary and involuntary movements of the body may

affect the contact; and, second, the body surface dissipates heat by evaporation as well as by radiation and conduction. This evaporation must not be interfered with by the application of the measuring instrument.

Different methods of applying thermocouples to the body surface have been suggested and used by different investigators. Among these, the most common are the direct application of the thermocouple to the skin without covering. For curved sections of the body, this may be accomplished by drawing the junction firmly against the surface by the leads. For other locations, the junction and a portion of the leads on either side may be held in contact with the surface by some form of web. In either case the temperature indicated will vary with the firmness of the contact—that is, whether the junction and the leads barely touch the surface, or whether they are pressed against the surface so as to be partly imbedded below the normal level of the surface.

Another method of applying the thermocouple is to attach it to some form of pad varying from ¼-in. to 1-in. in diameter. A cork pad suggests itself as the most advantageous type since it has low heat capacity and high resistance to heat flow. This cork pad application has definite advantages over the application of the bare couple. Any thermocouple, particularly when made of fine wire, is subject to breakage near the junction. Applying the couple attached as an intergral part of a cork or other pad greatly reduces the probability of breakage. The instrument, including the cork and the couple, can be applied to any part of the skin and held in place with uniform pressure with much less difficulty. Also, this application, for reasons discussed, gives more constant and consistent results.

One important factor in observing skin temperature is the lag or lapse of time between the application of the instrument and the final equilibrium temperature observed. This depends upon the heat capacity and resistance to heat flow of the entire assembly of the junction and anything else in contact with it. A bare couple, made of fine wire, attached directly to the surface should, obviously, give the least lag.

It will be readily seen that the application of the bare thermocouple to the skin offers a minimum of interference to the normal heat loss from the skin at the point where the measurement is to be made, while the application of a pad offers greater interference. If cork were applied to any considerable area of the body surface, the resultant skin temperature would approach blood, or the normal body temperature of 98.6 F. The closeness of this approach would depend upon the area covered and the thickness of the insulation used. An attempt is made to overcome this effect by limiting the area of the skin covered. Obviously, the smaller the area covered, the less the effect. Only experimentation can determine the magnitude of the resulting error accompanying the use of an instrument of this type.

In order to shed light on the relationship between the two methods of application and variation in the tightness of the application of the bare couple, a few observations were made which are plotted in Fig. 4. Thermocouple No. 1, after being in equilibrium with the air at about 76 F for a period of 5 min, was quickly applied to the skin with the usual degree of tightness in all such observations made at the Research Laboratory. A little more than half of the diameter of the No. 36 wire and junction was pressed below the normal level of the skin surface. It will be noted that this application gave complete equilibrium in about 1½ min and a constant indicated skin temperature thereafter. It should be observed that in these applications, the subject refrained more than normally from movements of his body or the muscles or skin of his neck. These movements would naturally have offered greater disturbance to the temperatures indicated. At the same instant, a second thermocouple was applied to the skin of the neck of the same subject at a point about an inch distant from the first couple. The junction and leads of this couple were drawn more tightly so that the entire cross-section of the wire and junction was depressed approximately one-half millimeter below the normal surface. This couple indicated a skin temperature approximately ½ deg higher than the first couple and came to equilibrium in approximately the same time.

After couple No. 2 had reached equilibrium with the

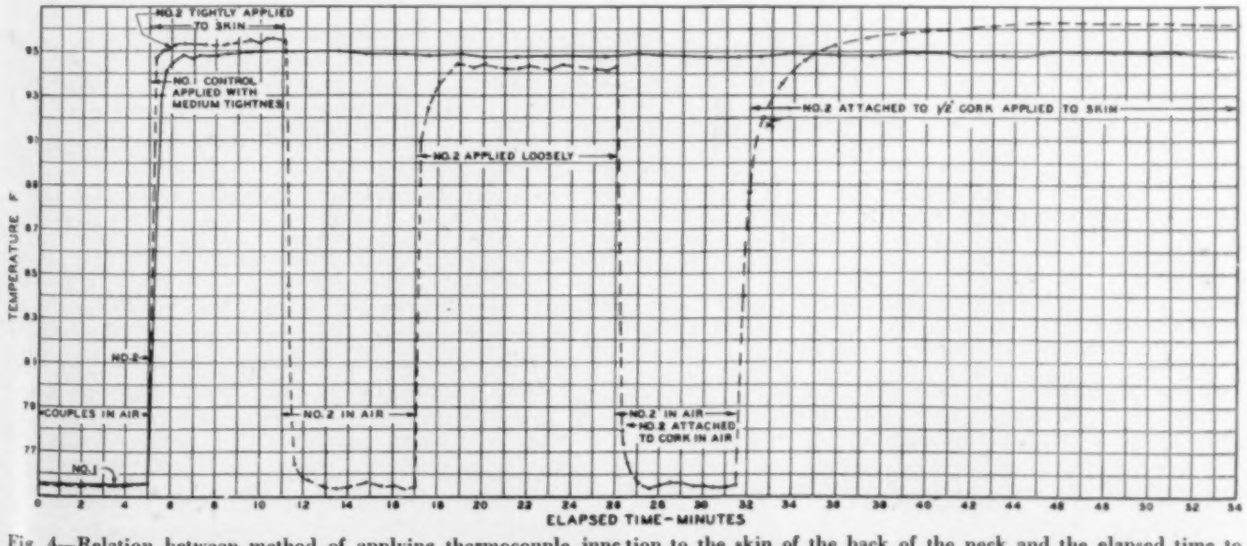


Fig. 4—Relation between method of applying thermocouple junction to the skin of the back of the neck and the elapsed time to reach equilibrium, the observed skin temperature and the constancy of the observed reading

skin, it was placed in the air long enough to become in equilibrium with it at an elapsed time of 17 min. Then it was again applied to the skin of the neck just tight enough to make sure that it was in contact. A little greater time is indicated for equilibrium to be reached, and a temperature about ½ deg lower is indicated. These three applications of the bare couple indicate variations which may be obtained under such conditions. One cannot, with certainty, assume that any one of the three is accurate. For all observations here reported, the Laboratory used the couple applied with medium tightness. The couples were made of No. 36 B.&S. gage copper-constantan wires, double silk covered. The wires were bare of insulation for a distance of approximately ⅛-in. either side of the junction. Beyond this distance, the insulation was firmly attached to the wire with wax.

At an elapsed time of 31½ min, couple No. 2 was attached to a cork ½ in. in diameter and about ⅜-in. thick. While both couple and cork were in equilibrium with the air, they were placed against the neck. Although the indicated temperature rose rapidly, it did not reach equilibrium until 10 or 12 min had elapsed. The indicated temperature was from 1 to 1¼ deg higher than that indicated by the bare couple applied with medium tightness. The reason for the greater lag is obviously due to the greater heat capacity of the cork and its initial temperature. The reason for the continued rise in the indicated temperature above that given by the bare couple was undoubtedly the interference with normal heat dissipation by the covered portion of the skin.

Observations of skin temperature with different methods of application indicate that a greater lag and a slightly higher temperature is obtained with a cork covered couple. On the other hand, a somewhat more constant temperature is indicated. Undoubtedly normal movements of the subject and chance variations in method of application have less effect. In some types of work ease of making observations and constancy of results may be more important than a small error in the indicated temperature. Furthermore, it should be observed that the thermocouple applied with cork was previously in equilibrium with the air at a considerably lower temperature. A more satisfactory performance would result if, while the instrument is not in use, it were kept in equilibrium with the temperature not widely different from the skin surface to be measured.

On the other hand, the bare couple, when properly applied, undoubtedly gives more nearly the true temperature. Much greater care must be exercised in applying it in order to avoid breakage and, if consistent results are desired, the junction should always be applied with the same degree of tightness. These differences would indicate that the laboratory practice of applying bare couples gives better results for research where the true temperature and its effect on heat transfer are important, and where greater skill on the part of the observer can be exercised in making the application. For routine observations, in places where variations in skin temperature of some considerable magnitude are the only objectives, the use of thermocouples applied to a cork pad may be equally satisfactory and more convenient.

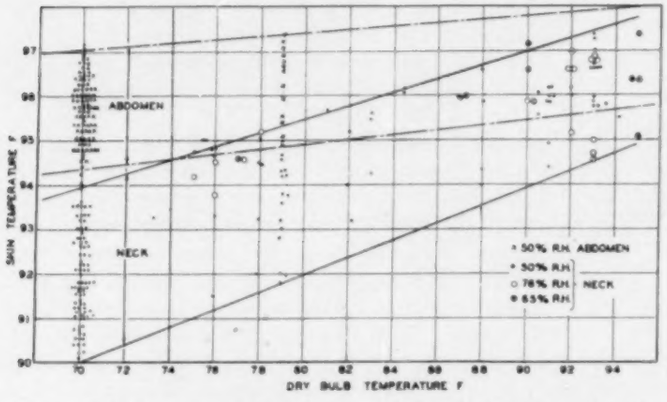


Fig. 5—Relation between dry-bulb temperature and the observed skin temperature of the neck and abdomen

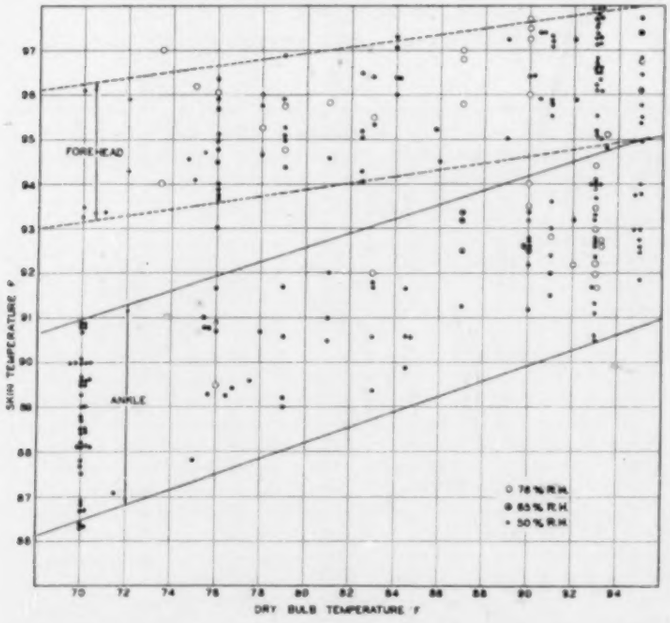


Fig. 6—Relation between dry-bulb temperature and the observed skin temperature of the ankle and the forehead

Normal Skin Temperatures

There has been some speculation and difference of opinion concerning the normal skin temperature of persons under different conditions of activity and in different atmospheric conditions. While this was not the purpose of the investigation, the many observations of skin temperature in equilibrium with given atmospheric conditions make it possible to shed additional light on this subject which may be of some interest. Figs. 5 and 6 give skin temperatures in equilibrium with air at different dry bulb temperatures and usually with 50 per cent relative humidity. A few observations were made at other relative humidities as indicated. The data show a spread of from 4 to 4½ deg for the skin temperature of the ankle and from 3 to 4 deg for the skin temperature of the neck at various air conditions. Curves are drawn giving approximately the limits of the spread. In all cases there is an

apparent rise in skin temperature with higher dry-bulb temperature. It is probable that more intensive study of this subject would indicate curves other than straight lines, particularly in the region of change from sensible to insensible perspiration. For the ankle, neck, and abdomen, the greatest number of observations was made at 70 F dry-bulb and 50 per cent relative humidity. It is of interest to note the widely different temperatures observed for the different parts of the body. All of the observations were made on a total of 15 male persons ranging from 18 to 30 years of age. It is not the purpose of this paper to discuss the physiological significance of these observations.

Drafts

The observations on the effect of drafts were made on 10 male subjects ranging from 18 to 30 years. For each test the subject remained in one of the psychrometric chambers, kept at the desired constant temperature and humidity, for at least a half-hour before observations were made.

In making the study it soon became apparent that more constant results could be obtained by observing the indicated change in skin temperature of any part of the body from a condition having substantially zero velocity, rather than from the normal air velocity pertaining in the psychrometric room. This was because of the small variations in air velocities in the psychrometric room set up by movements of observers and other uncontrolled factors which gave somewhat variable skin temperatures. Conditions of practically zero velocity were realized, for neck observations, by seating the subject, during the one-half hour preliminary period, with his neck close to the hood through which the impinging draft condition was later applied. No air movement was supplied in this period. Thus, undoubtedly, a normal skin temperature, lower than that which would have prevailed if this hood had not been used, was obtained.

Considerable difficulty was experienced in the preliminary study in getting consistent results on the ankle until it was observed that, almost without exception, there was a drop in ankle skin temperature after the subject sat down in the psychrometric room. This was corrected by applying thermocouples to the skin of both ankles so as to measure the change in the affected ankle from that of the ankle not affected by the draft. In order to eliminate further the uncontrolled effect of the variable air velocity in the room, the subject's two feet were placed in separate, corrugated paper shipping boxes. Preliminary observations were made on both ankles under these conditions. At the close of the time of the preliminary observations, air of the temperature and velocity constituting the draft to be studied was directed against one of the ankles, while the other remained in the substantially still air conditions pertaining in the other box. Therefore, all drops in skin temperature and feeling of discomfort due to the draft on the affected part are given in reference to substantially still air at 70 F and 50 per cent relative humidity, rather than to the reaction in room air of normal velocity as observed by the Kata-thermometer.

Fig. 7 shows the results of a single test on the neck of a subject who had previously spent some time in the psychrometric room with the back of his neck in close proximity to the hood as indicated in Fig. 3. Prior to zero elapsed time, the psychrometric chamber was held at 70 F dry-bulb and 50 per cent relative humidity, or 66 F effective temperature. The conditions in contact with the back of the neck were, therefore, approximately the same excepting that the air was still. Under these conditions, the neck temperature throughout the 20 min preliminary period, varied very little from 91 F,

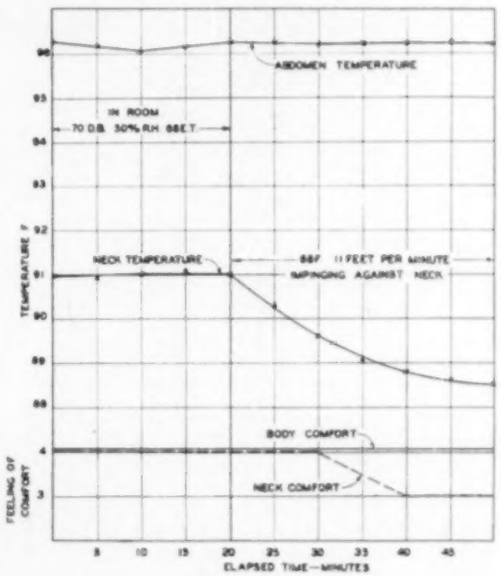


Fig. 7—Relation between elapsed time of exposure of neck to a draft of 68F dry-bulb temperature and air velocity of 11 fpm and the resulting skin temperature and feeling of comfort of the exposed part

until at an elapsed time of 20 min air at 68 F and a velocity of 11 fpm was directed against the back of the neck through the hood. The skin temperature of the neck dropped progressively and continuously for the next 30 min. The shape of the curve would indicate that a new equilibrium temperature would be reached somewhere near 88 F. For comparison of results, the change in skin temperature during the first 30 min was used in all cases. In this neck test there was a drop in skin temperature of 2½ deg in 30 min as a result of a draft of 68 F air at 11 fpm velocity below that pertaining for 70 F air at substantially no velocity.

Seventy similar tests were made on the neck with draft temperatures ranging from 70 to 65 F and velocities ranging from 0 to 110 fpm. The drop in skin temperatures for the different tests is plotted against the draft velocity, for the different draft temperatures, in Fig. 8. While there is considerable variation in the constancy of the test points, they do indicate a rather definite series of curves which show increased drop in skin temperatures with increase in velocity and with decrease in the draft temperature.

A single test on the ankle and the results of a large series of such tests are plotted in Figs. 9 and 10, respectively. It will be observed that after 30 min exposure the affected ankle temperature dropped slightly more than 2 deg below that of the ankle not affected by the draft. Within 40 min the drop in temperature reached over 2½ deg. The feeling of warmth indicated

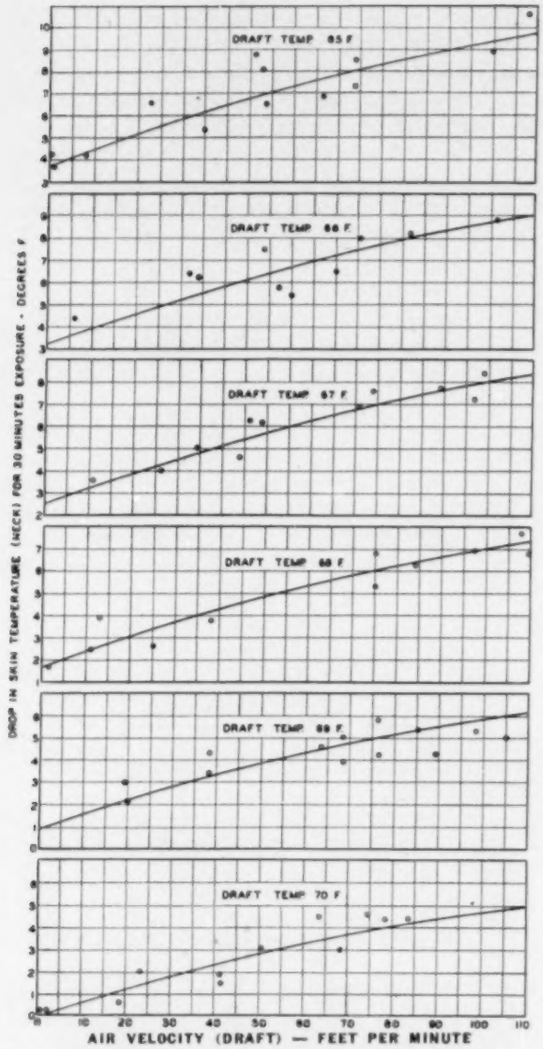

DRAFT TEMP. 65 F
DRAFT TEMP. 66 F
DRAFT TEMP. 67 F
DRAFT TEMP. 68 F
DRAFT TEMP. 69 F
DRAFT TEMP. 70 F
DROP IN SKIN TEMPERATURE (NECK) FOR 30 MINUTES EXPOSURE - DEGREES F
AIR VELOCITY (DRAFT) — FEET PER MINUTE

Fig. 8—Relation between air velocity (draft) and drop in normal skin temperature of the neck with a 30 min exposure for various draft temperatures. Persons seated at rest in a room at 70 dry-bulb and 50 per cent relative humidity (66 E.T.)

was *4* or *ideal* in the still air, and also in the draft until 27½ min elapsed. Within another 5 min *3* or *comfortably cool* was recorded.

The results of 71 tests on the ankle, given in Fig. 10, show about the same variations from the curves drawn, as for the neck. The two sets of curves have approximately the same characteristics, with a somewhat less drop in ankle temperature for a given draft condition than for the neck.

The relation between the drop in skin temperature and the per cent of comfort observations is plotted in Fig. 11 for the neck, and in Fig. 12 for the ankle. The same arbitrary index of comfort used in earlier laboratory studies was applied in this investigation. According to this index *4* designates *ideally comfortable, 3—comfortably cool, 5—comfortably warm, 2—cool, 6—warm, 1—decidedly cold* and *7—hot*. Greater difficulty was experienced in getting the subjects to properly interpret and evaluate these gradations of feeling in this study than was the case in earlier work where the entire body was subjected to changes in air conditions. It was particularly hard to distinguish between feelings *3* and *4*, and it is the opinion of those connected with the study that both indications of *3* and *4* should in most instances be construed as representing conditions which are not measurably objectionable. Besides the usual curves for comfort *1, 2, 3,* and *4* in Fig. 12, there is an additional curve representing the combined votes of *3* and *4*.

The relation between the draft velocity and the percentage of comfort (*3* and *4*) for different draft temperatures applied to the neck and ankle is given in Figs. 13 and 14, respectively. Figs. 15 and 16 give the relation between draft temperature and draft velocity for

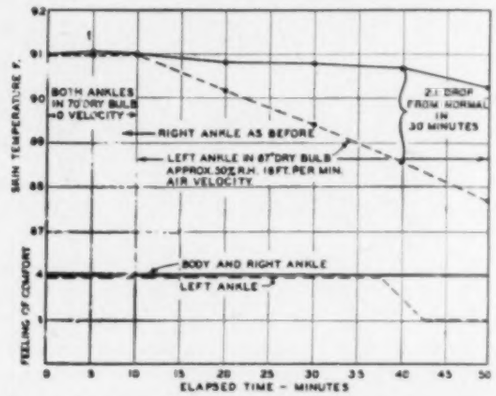

SKIN TEMPERATURE F.
FEELING OF COMFORT
BOTH ANKLES IN 70 DRY BULB 0 VELOCITY
RIGHT ANKLE AS BEFORE
LEFT ANKLE IN 67° DRY BULB APPROX. 50% R.H. 16 FT. PER MIN. AIR VELOCITY
2.1 DROP FROM NORMAL IN 30 MINUTES
BODY AND RIGHT ANKLE
LEFT ANKLE
ELAPSED TIME - MINUTES

Fig. 9—Relation between elapsed time of exposure of the ankle to a draft of 67F dry-bulb temperature and air velocity of 16 fpm, and the resulting skin temperature and feeling of comfort of the exposed ankle

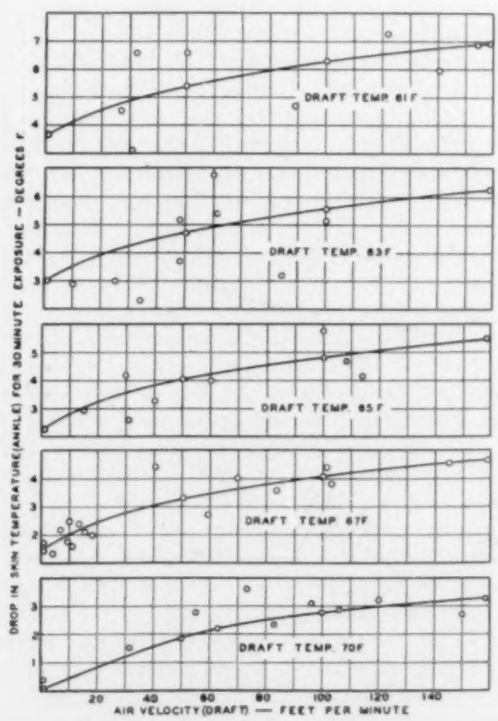

DRAFT TEMP. 61 F
DRAFT TEMP. 63 F
DRAFT TEMP. 65 F
DRAFT TEMP. 67 F
DRAFT TEMP. 70 F
DROP IN SKIN TEMPERATURE (ANKLE) FOR 30 MINUTE EXPOSURE — DEGREES F
AIR VELOCITY (DRAFT) — FEET PER MINUTE

Fig. 10—Relation between air velocity (draft) and drop in normal skin temperature of the ankle with a 30 min exposure for various draft temperatures. Persons seated at rest in a room at 70 F dry-bulb and 50 per cent relative humidity (66 E.T.)

Table 1—Objectionable Draft Conditions Based Upon 90 Per Cent Comfort (3 and 4)

Neck		Ankle	
Dry-Bulb Temp. F	Velocity Fpm	Dry-Bulb Temp. F	Velocity Fpm
70	60	70	165
69	40	69	111
68	24.5	68	77
67	12.5	67	55
66	0	66	37
		65	24
		64	13
		63	6
		62	0

different drops in skin temperature for the neck and ankle, respectively.

If a 90 per cent indication of comfort, *3* and *4*, is accepted as the borderline between unobjectionable and objectionable drafts, then it will be noted from Figs. 11 and 12 that drops in skin temperature of 3.3 and 3.4 represent the borderline between satisfactory and unsatisfactory conditions for the neck and ankle, respectively. Thus the curve for these temperature drops in Figs. 15 and 16 (indicated by the heavy broken-line) represents conditions of temperature and velocity at and above which objectionable drafts occur.

Based upon the heavy broken line curves in Figs. 15 and 16, the combinations of temperatures and velocities which constitute unsatisfactory draft conditions, are given in Table 1 for the neck and ankle. Accepting these values, it will be noted that air having a temperature of 70 F and velocity of 60 fpm directed against the neck becomes intolerable. On the other hand, if the

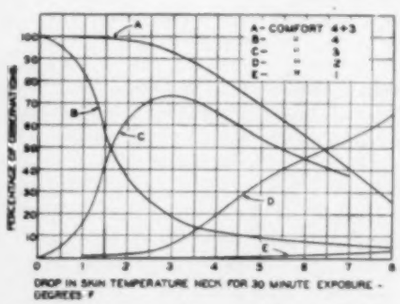


Fig. 11—Relation between drop in skin temperature of the neck and per cent of different degrees of comfort

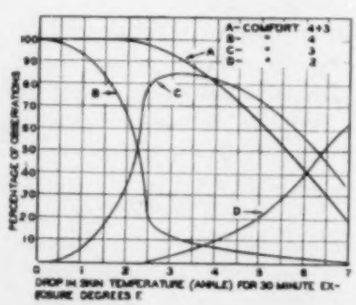


Fig. 12—Relation between drop in skin temperature of the ankle and per cent of different degrees of comfort

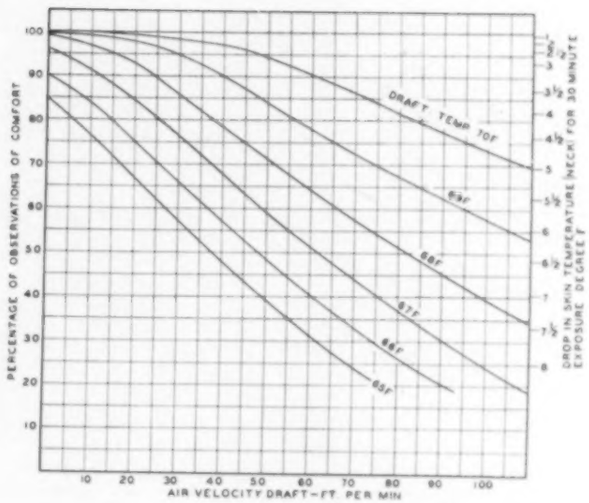


Fig. 13—Relation between air velocity (draft) and per cent of observations of comfort (left hand scale) and drop in neck skin temperature (right hand scale) after 30 min exposure for different draft temperatures

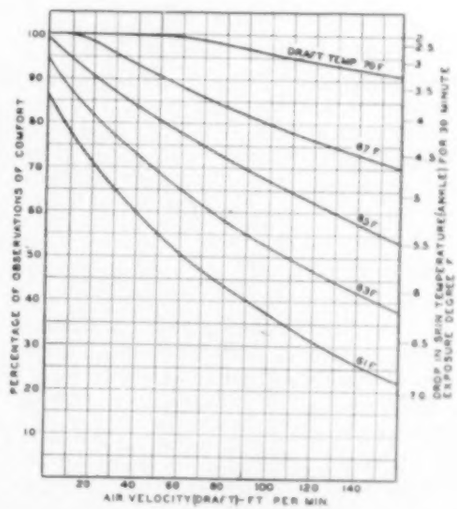

DRAFT TEMP 70 F
67 F
65 F
63 F
61 F
PERCENTAGE OF OBSERVATIONS OF COMFORT
DROP IN SKIN TEMPERATURE (ANKLE) FOR 30 MINUTE EXPOSURE DEGREE F
AIR VELOCITY (DRAFT) - FT PER MIN

Fig. 14—Relation between air velocity (draft) and per cent of observations of comfort (left hand scale) and drop in ankle skin temperature (right hand scale) after 30 min exposure for different draft temperatures

velocity is zero, an air temperature of 66 F is equally intolerable. An air velocity of 25 fpm, as measured by a Kata-thermometer, becomes an objectionable draft when the air temperature drops to 68 F. This is in agreement with the usually accepted tolerance of ± 2 F in air conditioning practice.

The velocity of 165 fpm, which was indicated as objectionable when applied to the ankle of a subject sitting in 70 F air, seems excessively high. It should be kept in mind, however, that a person's ankle is not ordinarily subjected to 70 F air, because of the usual temperature gradient in winter air conditioned rooms. A 70 F temperature at the 3-ft level is usually associated with a floor-line temperature of 65 F. Table 1 indicates that a 65 F floor-line temperature and a 24 fpm velocity are uncomfortable, while with no air velocity a floor-line temperature of 62 F becomes equally uncomfortable.

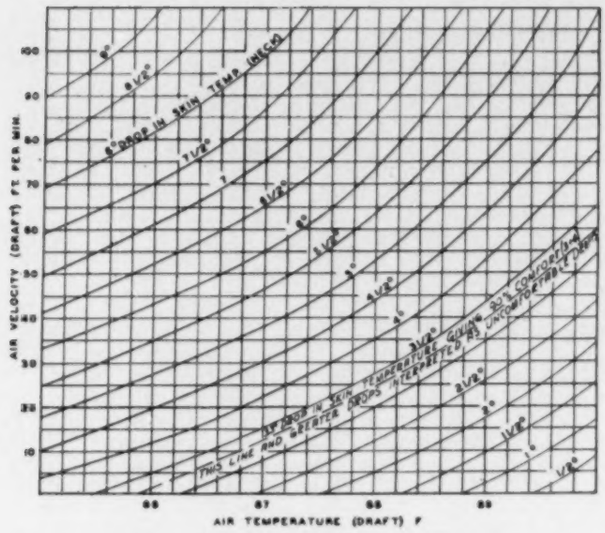


Fig. 15—Relation between the temperature and velocity of a draft and the drop in skin temperature of the neck. Skin temperature drop of 3.3 deg and greater interpreted as objectionable draft

The data given in Figs. 13 to 16 indicate a drop in skin temperature of 0.83 deg for the neck and 0.51 deg for the ankle for a 1 deg drop in dry-bulb temperature of the air when these parts of the body alone are affected by a draft. It is of interest to compare these relationships with those pertaining when the entire body is subjected to changes in temperature. From Figs. 5 and 6 the slope of the neck and ankle temperature curves indicates changes in neck and ankle skin temperatures of 0.18 and 1.17 respectively, per degree temperature change in room air affecting the entire body. At first thought it is not easy to adjust these differences. A complete knowledge of conditions would probably reconcile them on the basis that when the temperature of the air affecting the entire surface of the body is changed, the body compensates by a change in the availability of perspiration for evaporation. However, when a cooler air temperature under draft conditions affects only a limited surface area, this compensation is not provided; at least, not to the same extent.

It is of interest to note the relative effect of air temperature and air velocity in producing a given cooling effect within the temperature range of 65 to 70 F and the velocity ranges of 0 to 50 fpm. Figs. 15 and 16 show that a change in air velocity of about 15 fpm is equivalent to a change in air temperature of 1 deg.

Conclusions

1. The study shows that there is a definite relation between the drop in the skin temperature of the ankle and neck and the temperature and velocity of a draft of air striking these parts. Data on these relationships and the resulting feeling of warmth are given for room condition of approximately 70 F and 50 per cent relative humidity only. For the same change in air temperature and velocity a greater change in skin temperature is indicated for the neck.

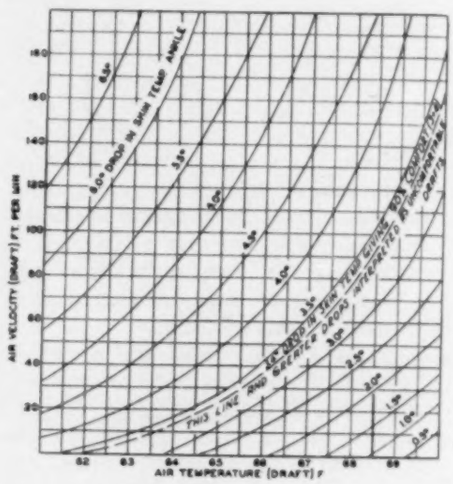


Fig. 16—Relation between the temperature and velocity of a draft and the drop in skin temperature of the ankle. Skin temperature drop of 3.4 deg and greater interpreted as objectionable draft

2. This paper, accepting a recommended percentage of 90.0 for comfort indices of *3* and *4* combined, gives minimum limits in temperature and maximum limits of velocity at and beyond which intolerable drafts occur for neck and ankle.

3. A considerably higher draft velocity and lower draft temperature are indicated as possible for the ankle than for the neck. This difference is probably due to the lower temperature to which the ankle is accustomed in modern heating practice.

Symposium on Drying and Air Conditioning

The Fourth Chemical Engineering Symposium of the Division of Industrial and Engineering Chemistry of the *American Chemical Society* occurred on December 27 and 28 at the University of Pennsylvania at Philadelphia, Pa. Among the more than 200 persons attending were A. E. Stacey, New York, and F. R. Bichowsky, Ann Arbor, Mich.

The program included the following papers which were presented: Fundamentals of Drying and Air Conditioning, by E. R. Gilliland; An Equilibrium Pressure Chart for Two Components, One Volatile, by A. B. Newman; The Drying of Air and Commercial Gases with Activated Alumina, by R. B. Derr; Accurate Determination of Dew Point, by A. W. Hixson and G. E. White; The Evaporation of Water into Quiet Air from a One Foot Diameter Surface, by B. F. Sharpley and L.-M. K. Boelter; Use of Lithium Chloride in Drying and Air Conditioning, (a) Thermodynamic Properties of Lithium Chloride Solutions, (b) Application to Adiabatic Drying, by F. R. Bichowsky; Rate of Evaporation from a Free Water Surface to a Transverse Air Stream, by M. C. Molstad; The Drying of Materials in Trays—Evaporation of Surface Moisture, by C. B. Shepherd, C. Hadlock and R. C. Brewer; Comparison of Rate of Evaporation of Moisture from a Wet Material and from a Free Water Surface, by A. E. Stacey, Jr.; Vacuum Drying of Solids, by R. C. Ernst; and Technical and Economic Aspects of Spray Drying, by B. B. Fogler and R. V. Kleinschmidt.

Correlating Thermal Research

As a part of the efforts of the A. S. H. V. E. Research Laboratory to correlate research in thermal engineering carried on by the many institutions engaged in such work, and to disseminate the published results of such studies together with other reports of progress in the field, and in order to make this information available to the membership of the Society, there will be published monthly on this page a limited number of brief abstracts of articles which it is believed will be of interest to all concerned. For more complete lists address the Librarian, A. S. H. V. E. Research Laboratory, U. S. Bureau of Mines Experiment Station, Pittsburgh, Pa.

F. C. Houghten, *Director.*

W. L. Fleisher, *Chairman.*
COMMITTEE ON RESEARCH

• Direct Reading Effective Temperature Indicator, by J. R. Parson, *Refrigerating Engineering*, Vol. 34, No. 6, Dec. 1937, pp. 359-363. Discussion of effective temperature chart developed by A.S.H.V.E., also Parsons Comfort chart on which effective temperature is read directly from dry-bulb and relative humidity values. 50 per cent relative humidity line is basis of chart. Description of Bailey-Parsons Therhumiter or effective temperature indicator.

• Principles of Exhaust Hood Design, by John L. Alden, *Heating and Ventilating*, Vol. 34, No. 10, Oct. 1937, pp. 56-60. Description of design and use of the three types of exhaust hoods, machine hoods, top or canopy hoods, and booths or enclosures.

• Importance of Air Filters, *Automatic Heat and Air Conditioning*, Vol. 8, No. 11, Nov. 1937, pp. 50-51. Discussion and diagrams of air filters in air conditioning systems. Description of dry mat and rotating oil filters, with advantages of each.

• Factors in the Fabrication of Sheet Metal Ducts, by O. W. Kothe, *Automatic Heat and Air Conditioning*, Vol. 8, No. 12, Dec. 1937, pp. 40-44. Discussion of and sketches showing in detail, fabrication methods for sheet metal ducts. Streamline duct curves and joints. Also complete air conditioning system showing filters, fans, humidity pans, compressors, and duct distributing system.

• Air Conditioning Theory and Practice, by Arthur M. Branch and Rolf Toensfeldt, *Building and Building Management*, Mar. 1937. Technical with many details. Discussion of human heat regulation by perspiration, beginning at 63 F with maximum effect at 85 F. Air Conditioning aims to hold temperature to produce most comfortable conditions.

• Modern Principles of Ventilation and Heating, by T. Bedford. Three lectures given under the Heath Clark bequest to the National Institute of Industrial Psychology. 94 pages. Pub. by H. K. Lewis & Co. Principles of ventilation and heating with summaries of current views on the subject. Results of a recent large-scale factory investigation to find most comfortable conditions of warmth for persons doing sedentary work. Effect of atmospheric conditions on many aspects of industrial efficiency.

• Glass and Concrete Construction, by G. Schieb, *Oesterr Glaserzty*, Vol. 2, pp. 86-89, 1937. The method of construction of a small dome carrying round glass inserts is illustrated photographically.

• Condensation in Attics and Walls, by L. V. Teesdale, Forest Service, U. S. Dept. of Agriculture, *Automatic Heat and Air Conditioning*, Vol. 8, No. 12, Dec. 1937, pp. 46-48. Discussion of accumulation of water vapor in walls or attic spaces. Increased relative humidity in better homes, intended to be better suited to health, is responsible for new condensation problems. Water vapor moves by diffusion from zones of higher vapor pressure to zones of lower vapor pressure. It moves readily through plain plaster but is retarded by paint or other surface coatings. In cold weather vapor may pass through insulation and freeze on inside of roof boards or siding. Later ice melts and the water leaks through ceiling making roof appear to leak. Proper attic ventilation will prevent roof condensation.

• Ventilation of Factories and Workshops. Home Office Welfare Pamphlet No. 5, 3rd edition, pp. 52, H. M. Stationery Office, London, 1937. Price 1s. Paper deals briefly with: 1. Conditions necessary for satisfactory ventilation and ventilation standards; 2. Arrangement for natural ventilation; 3. Various types of fans; 4. Different systems of mechanical ventilation; 5. Note on heating in combination with ventilation; and 6. Instruments for determining environmental conditions.

• Air Conditioning in the Tropics, by C. P. Mom, *De Ingen. in Nederland Indie*, Vol. 6, 1937. Description of development of comfort zones, for men at rest dressed in ordinary tropical clothes. Conditions in Java and India are found almost identical. Data on average outdoor temperature and humidities for last 75 years; also some extreme weather conditions. Optimum day and night temperatures suggested.

• Automatic Temperature Control. Pamphlet published by Thermocontrol Installations Co. Ltd., 68 Victoria Street, S.W.1, and the *London and Counties Coke Association*. Discussion of scope and advantages of automatic control of temperature, including instruments used, principles of operation and illustrations of various types of control, thermostatic control of solid fuel boilers, thermostatic control for various industrial processes.

• Methods for the Determination of Quartz in Industrial Dusts, by F. H. Goldman, U. S. Public Health Service, *Public Health Reports*, Vol. 52, No. 48, Nov. 26, 1937, pp. 1702-1712. Methods for determining quartz are in two classes, chemical and petrographic. Procedure in various tests.

• Air Conditioning, by Oscar Faber, *Engineering*, Vol. CXLIV, No. 3748, Nov. 12, 1937, pp. 556-558. Discussion of need of air conditioning for human comfort. Air conditioning treated in three phases: 1. Adjustment of temperature; 2. Adjustment of humidities; 3. Filtration. Diagram of essentials of an air conditioning plant in which the relative position and arrangement of the various parts are indicated.

• Radiator Heat Systems for Two-Family House with Individual Air Conditioning, *Heating and Ventilating*, Vol. 34, No. 10, Oct. 1937, pp. 37-39. Simple forced hot water heating system supplies the whole building. Two air conditioners located in basement, one for each apartment, humidification, filtration and air circulation are provided.

• Hot Water Supply, *Domestic Engineering*, Vol. 150, No. 5, Nov. 1937. How and why domestic hot water systems function according to basic principles. Diagrams of layout for various hot water supply systems in residences, showing use of air vents and circulators.

• Measuring Skin Temperature, by J. D. Hardy and G. F. Soderstrom, The Review of Scientific Instruments, Vol. 8, No. 11, Nov. 1937, pp. 419-22. Description of radiometer for measuring the radiation from the skin surface and an especially adapted thermocouple for measuring the internal body temperature. The whole instrument is portable, rugged and simple to operate. Readings are made directly in skin temperature and internal body temperature units. The device is arranged so that the radiation from the skin is compared to the radiation from a black body of known temperature.

• Prevention of Clinker on Industrial Stokers, by J. R. Darnell, *Power Plant Engineering*, Vol. XLI, No. 12, Dec. 1937, pp. 752-754. Discussion of prevention of and removal of clinker on industrial stokers when low grade coal is used. Fusion points of ash of various grades of bituminous coals vary from 1800 to 3000 F. Most mid-west coal has low ash fusion point, so precautions must be taken in burning such coal and in removing the ash. Use of chain or traveling grate suggested for stokers. Tempering of coal lessens formation of clinker.

NEWS OF LOCAL CHAPTERS

OFFICERS OF LOCAL CHAPTERS—1938

ATLANTA: *Organized,* 1937. *Headquarters,* Atlanta, Ga. *Meets,* First Tuesday. *President,* E. W. KLEIN, 152 Nassau St., N. W. *Secretary,* C. T. BAKER, 713 Glenn St., S. W.

CINCINNATI: *Organized,* 1932. *Headquarters,* Cincinnati, O. *Meets,* Second Tuesday in Month. *President,* I. B. HELBURN, 610 Chamber of Commerce Bldg. *Secretary,* H. E. SPROULL, 1005 American Bldg.

GOLDEN GATE: *Organized,* 1937. *Headquarters,* San Francisco, Calif. *Meets,* First Tuesday. *President,* B. M. WOODS, Univ. of Calif., Berkeley, Calif. *Secretary,* G. J. CUMMINGS, 113 Tenth St., Oakland, Calif.

ILLINOIS: *Organized* 1906. *Headquarters,* Chicago, Ill. *Meets,* Second Monday. *President,* S. I. ROTTMAYER, 407 S. Dearborn St. *Secretary,* C. E. PRICE, 6 N. Michigan Ave.

IOWA-NEBRASKA: *Organized,* 1937. *Headquarters,* Omaha, Neb. *President,* M. J. STEVENSON, 1643 South 20th St., Lincoln, Neb. *Secretary,* W. R. WHITE, 4339 Larimore Ave., Omaha, Neb.

KANSAS CITY: *Organized,* 1917. *Headquarters,* Kansas City, Mo. *Meets,* Second Monday in Month. *President,* A. H. SLUSS, 827 Mississippi Ave., Lawrence, Kan. *Secretary,* GUSTAV NOTTBERG, 914 Campbell St.

MANITOBA: *Organized,* 1935. *Headquarters,* Winnipeg, Man. *Meets,* Fourth Thursday. *President,* D. F. MICHIE, 492 Wardlaw Ave. *Secretary,* E. J. ARGUE, Ste. 23, Estelle Apts.

MASSACHUSETTS: *Organized,* 1912. *Headquarters,* Boston, Mass. *Meets,* Third Tuesday in Month. *President,* JAMES HOLT, Massachusetts Institute of Technology, Cambridge, Mass. *Secretary,* H. C. MOORE, 69 Massachusetts Ave., Cambridge, Mass.

MICHIGAN: *Organized,* 1916. *Headquarters,* Detroit, Mich. *Meets,* First Monday after the 10th of the Month. *President,* F. J. FEELY, 950 Trombley Rd., Grosse Pointe Pk. *Secretary,* G. H. TUTTLE, 2000 Second Ave.

WESTERN MICHIGAN: *Organized,* 1931. *Headquarters,* Grand Rapids, Mich. *Meets,* Second Monday in Month. *President,* W. W. BRADFIELD, 901 Michigan Trust Bldg. *Secretary,* S. W. TODD, JR., 309 Paris, S. E.

MINNESOTA: *Organized,* 1918. *Headquarters,* Minneapolis, Minn. *Meets,* Second Monday in Month. *President,* R. E. BACKSTROM, Room 1981, First Natl. Bank Bldg., St. Paul, Minn. *Secretary,* F. C. WINTERER, 836 Juno St., St. Paul, Minn.

MONTREAL. *Organized,* 1936. *Headquarters,* Montreal, Que. *Meets,* Third Monday. *President,* G. L. WIGGS, University Tower. *Secretary,* C. W. JOHNSON, 630 Dorchester St., W.

NEW YORK: *Organized,* 1911. *Headquarters,* New York, N. Y. *Meets,* Third Monday in Month. *President,* W. E. HEIBEL, 11 West 42nd St. *Secretary,* T. W. REYNOLDS, 100 Pinecrest Dr., Hastings-on-Hudson, N. Y.

WESTERN NEW YORK: *Organized,* 1919. *Headquarters,* Buffalo, N. Y. *Meets,* Second Monday in Month. *President,* B. C. CANDEE, 19 Tremont Ave., Kenmore, N. Y. *Secretary,* W. R. HEATH, 119 Wingate Ave.

NORTHERN OHIO: *Organized,* 1916. *Headquarters,* Cleveland, O. *Meets,* Second Thursday in Month. *President,* PHILIP COHEN, 401 East Ohio Gas Bldg. *Secretary,* C. A. MCKEEMAN, Case School of Applied Science.

OKLAHOMA: *Organized,* 1935. *Headquarters,* Oklahoma City, Okla. *Meets,* Second Monday. *President,* E. F. DAWSON, University of Oklahoma, Norman, Okla. *Secretary,* E. W. GRAY, Box 1498, Oklahoma City, Okla.

ONTARIO: *Organized,* 1922. *Headquarters,* Toronto, Ont. *Meets,* First Monday in Month. *President,* G. A. PLAYFAIR, 113 Simcoe St. *Secretary,* H. R. ROTH, 57 Bloor St. W.

PACIFIC NORTHWEST: *Organized,* 1928. *Headquarters,* Seattle, Wash. *Meets,* Second Tuesday in Month. *President,* W. W. COX, 326 Columbia St. *Secretary,* M. N. MUSGRAVE, 314-9th Ave., N.

PHILADELPHIA: *Organized,* 1916. *Headquarters,* Philadelphia, Pa. *Meets,* Second Thursday in Month. *President,* L. P. HYNES, 240 Cherry St. *Secretary,* C. B. EASTMAN, 530 Brookview Lane, Brookline, Upper Darby, Pa.

PITTSBURGH: *Organized,* 1919. *Headquarters,* Pittsburgh, Pa. *Meets,* Second Monday in Month. *President,* J. F. COLLINS, JR., 435 Sixth Ave. *Secretary,* T. F. ROCKWELL, Carnegie Inst. Tech.

ST. LOUIS: *Organized,* 1918. *Headquarters,* St. Louis, Mo. *Meets,* First Tuesday in Month. *President,* G. W. F. MYERS, 3947 W. Pine Blvd. *Secretary,* D. J. FAGIN, 1344 Woodruff Ave.

SOUTHERN CALIFORNIA: *Organized,* 1930. *Headquarters,* Los Angeles, Calif. *Meets,* Second Tuesday in Month. *President,* E. H. KENDALL, 1978 S. Los Angeles St. *Secretary,* J. F. PARK, 1234 South Grand.

TEXAS: *Organized,* 1936. *Headquarters,* College Station, Texas. *President,* R. F. TAYLOR, 909 Banker's Mortgage Bldg., Houston, Tex. *Secretary,* W. H. BADGETT, Texas Engrg. Experiment Station, College Station, Tex.

WASHINGTON, D. C.: *Organized,* 1935. *Headquarters,* Washington, D. C. *Meets,* Second Wednesday in Month. *President,* L. OUBUSOFF, 411 Tenth St., N. W. *Secretary,* L. F. NORDINE, Room 203, 734 Jackson Pl., N. W.

WISCONSIN: *Organized,* 1922. *Headquarters,* Milwaukee, Wis. *Meets,* Third Monday in Month. *President,* J. H. VOLK, 1906 W. St. Paul Ave. *Secretary,* H. C. FRENTZEL, 3000 W. Montana St.

Talks on Coal, Oil and Gas Given at Philadelphia Meeting

December 9, 1937. The meeting of the Philadelphia Chapter was called to order at 8:00 p. m. by Pres. L. P. Hynes. The minutes of the previous meeting and the treasurer's report were read and approved.

The report was made by H. H. Mather, Membership Committee, that applications were being filed by 9 for membership in the local Chapter, which would make a total of 25 new members for the season.

A resolution was passed instructing Secy. C. B. Eastman to write a letter of condolence to H. H. Erickson in connection with the passing of his mother.

M. F. Blankin announced plans for the January meeting.

President Hynes appointed R. C. Bolsinger and A. H. MacDade to prepare a resolution expressing the deep regret of the Chapter in the passing of C. E. Dobbs.

F. D. Mensing and J. D. Cassell spoke in opposition to the proposed amendment to the By-Laws of the A.S.H.V.E. with reference to Article B-XI, Section 1.

Excellent talks were given by A. J. Johnson, Anthracite Laboratories, Primos, Pa., on Coal; by T. H. Smoot, Anchor Fence Co., Baltimore, Md., on Oil; and by F. R. Trembly, Jr., Philadelphia Gas Works Co. on Gas. Active discussion followed these talks which resulted in such an interesting meeting that adjournment did not occur until 10:30 p.m.

Joint Meeting Held by Ontario Chapter

January 10, 1938. A joint meeting was held at the Royal York Hotel, Toronto of the *A.S.M.E.* local Section, the Engineering Institute of Canada, and the local Chapter of the A.S.H.V.E. Dinner was served at 6:30 p.m. and 168 members and guests were present. W. G. McIntosh of the *A.S.M.E.* acted as chairman.

H. B. Jenney announced the untimely death of A. K. Purdy, who had been a member of the A.S.H.V.E. for many years.

E. H. Gurney introduced the principal speaker of the evening, Prof. E. A. Allcut, who gave an illustrated talk on Heat Insu-

lating Materials and Their Properties. Dean Mitchell of the University of Toronto thanked the speaker on behalf of those present for the instructive talk which he presented.

F. E. Ellis, Prof. R. W. Angus, and C. Tasker spoke for the Engineering Institute of Canada, the *A.S.M.E.*, and the A.S.H.V.E., respectively, on the advantages of holding a joint meeting. It was voted to hold such a gathering again next year.

H. S. Moore extended an invitation to all those present to attend the 44th Annual Meeting of the A.S.H.V.E. and the Heating and Ventilating Exposition in New York, January 24-28, before the meeting adjourned.

Unit Heaters and Coolers Discussed at Atlanta Meeting

January 11, 1938. Pres. E. W. Klein presided at the meeting of Atlanta Chapter held at the Biltmore Hotel. The meeting was preceded by dinner and proved to be an enjoyable affair with a good attendance.

The regular business included the reading of the minutes of the previous meeting by Secy. C. T. Baker and a letter from Secy. A. V. Hutchinson of the A.S.H.V.E., advising the Chapter of the election of four new members from the Atlanta district.

President Klein then introduced W. A. Evans, district manager, Aerofin Corp., who gave an informal talk on Unit Heaters and Coolers, which included a discussion of several types available.

In his talk, Mr. Evans gave pertinent information concerning the operation and performance of the apparatus, which, in connection with the discussion that followed, proved of interest and value to all. It was the opinion of members that the meeting was one of the most interesting and satisfactory in the history of the Chapter.

Hotel Air Conditioning Subject of Montreal Meeting

December 13, 1937. Fifty members and guests attended the regular meeting of the Montreal Chapter at the Mount Royal Hotel. Following dinner the meeting was called to order at 8:00 p.m. by Vice-Pres. F. J. Friedman. After roll call and the introduction of guests, a letter of welcome from the hotel manager was read by Secy. C. W. Johnson.

G. L. Wiggs, president of the Chapter and consulting engineer, was the speaker of the evening. He was introduced by W. U. Hughes and discussed the subject of Air Conditioning in the Mount Royal Hotel. Mr. Wiggs gave a detailed description of the air conditioning systems in the Normandy Roof, Piccadilly Club, and refrigeration equipment used in connection with these two systems. The talk was illustrated by slides and diagrams and a vote of thanks to Mr. Wiggs was extended by Henri Gendron.

L. H. Laffoley made an announcement concerning the Annual Meeting of the A.S.H.V.E. and urged attendance by Montreal members.

At 9:50 p.m. the meeting adjourned and a tour of inspection of the hotel's air conditioning system was made.

Western New York Chapter Enjoys Christmas Party

December 20, 1937. Approximately 50 members and guests of the Western New York Chapter met at the University Club, Buffalo.

The Chapter voted to appoint Pres. B. C. Candee official delegate, as well as representative on the Society's Nominating Committee, with Vice-Pres. J. J. Landers as alternate, for the 44th Annual Meeting of the A.S.H.V.E.

A letter from the Northern Ohio Chapter requesting endorsement of Cleveland as the location for the 1940 meeting of the Society was read and approved.

A resolution was unanimously adopted extending the Chapter's heartiest congratulations to L. A. Harding upon his appointment as Commissioner of Public Works for the City of Buffalo and sending him best wishes for a successful administration.

President Candee announced tentative plans for a special car to the Society's 44th Annual Meeting in New York.

The meeting was then turned over to Joseph Davis, who was in charge of the annual Christmas party. Mr. Davis had arranged a Kicker's Handicap Bowling Tournament and Bridge. Refreshments were provided and an excellent time was reported by all.

Annual Banquet of Manitoba Chapter

December 11, 1937. Manitoba Chapter members and guests held their annual banquet in the Fort Garry Hotel, Winnipeg, with 24 members and guests present. Pres. D. F. Michie presided.

Grace was asked by J. B. Steele, and, following dinner, toasts were proposed to the King by President Michie, to the Society by R. L. Kent, and to the ladies and guests by I. McDonald. Mr. Steele replied to Mr. Kent, while Mrs. E. J. Argue responded to Mr. McDonald.

After an enjoyable evening of dancing, the party dispersed at 12:30 a. m.

Automatic Control of Refrigeration Subject of Illinois Chapter Meeting

December 13, 1937. The Illinois Chapter met at the Brevoort Hotel, Chicago, with 80 present at dinner preceding the regular order of business, which Pres. S. I. Rottmayer opened at 7:45 p. m.

S. M. Trumbo and A. L. Crump were appointed as tellers to count the ballots for the election of treasurer and member of the Board of Governors. The unanimous election of Tom Brown as treasurer and O. W. Armspach as member of the Board of Governors was declared.

Secy. C. E. Price read the minutes of the last meeting and E. M. Mittendorff introduced three new Chapter members, L. C. Mabley, Lloyd Howell and H. E. Ryerson. A visiting member from the Philadelphia Chapter, Mr. Fenn, was introduced and welcomed to the meeting.

A report of a special Chapter committee, consisting of R. E. Hattis, I. E. Brooke and J. R. Vernon, on a proposal to organize an Illinois Engineering Council was given by Mr. Hattis. Such a Council would consist of all professional engineering organizations in the Chicago district and would function on general legislative and other matters of common concern to all. Mr. Hattis stated that the idea was endorsed in principle by the representatives of the various engineering organizations attending the special meeting called for its consideration, and a committee was appointed to draft a prospectus and present details for further study by the different groups.

A. B. Schellenberg, Alco Valve Co., Inc., St. Louis, spoke on Automatic Control of Refrigeration. Because of the wide range of the subject, Mr. Schellenberg confined his discussion to those automatic control devices used to regulate the flow of the refrigerant and more specifically to the thermostatic expansion valve. He covered briefly the constant pressure expansion valve, the high pressure float valve, the low pressure float switch and magnetic stop valve, and then explained the thermostatic expansion valve in detail—its construction, its operation, the effect of evaporator design upon its operation and its particular control functions. Slides helped to illustrate these various points.

In operation for further study was a demonstrating glass evaporator, hooked up with a small compressor and fed by a thermostatic expansion valve. With this demonstrator it was possible to show exactly what happens inside an evaporator under many different operating conditions. Mr. Schellenberg's talk

was received with great interest, and appreciation for its thoroughness and informativeness was recorded in a rising vote of thanks.

Pittsburgh Chapter Elects Officers

December 13, 1937. Pres. M. L. Carr called the meeting to order at 8:20 p. m., following the Pittsburgh Chapter's annual dinner in the East Room of the Hotel Roosevelt.

Treas. J. E. McLean read the financial report, and P. A. Edwards, chairman of the Membership Committee, gave a résumé of the changes in membership of various grades which occurred during 1937. Mr. Edwards showed a net increase of eight for the Pittsburgh Chapter.

Secy. T. F. Rockwell read a letter from headquarter's office concerning the resumption of the admission fee beginning in 1938, and a letter from the Cleveland Chapter regarding the 1940 annual meeting of the Society.

R. J. J. Tennant repeated the report of the Nominating Committee. The nominations were closed and the Secretary was directed to cast a unanimous ballot for the candidates as nominated by the Committee. As a result, the following officers of the Pittsburgh Chapter were elected:

President—J. F. S. Collins, Jr.
Vice-President—R. A. Miller.
Secretary—T. F. Rockwell.
Treasurer—L. S. Maehling.
Board of Governors—M. L. Carr, *chairman;* E. H. Riesmeyer, Jr.; E. C. Smyers.

Mr. Carr, retiring president, reviewed the activities of the Chapter for 1937 and presented the Chapter with a book which he called Who's Who in Pittsburgh Chapter. Mr. Carr had prepared a questionnaire and had collected information from the active members and bound the sheets in a loose leaf cover so that the succeeding Chapter officers would have available an active file of the membership.

To induct the new president into office, Mr. Carr presented Mr. Collins with a gavel as a symbol of authority. President Collins made a few inaugural remarks and a rising vote of thanks was then given to Mr. Carr.

President Collins introduced Councilman George Evans as the guest speaker. Mr. Evans, who is also head of the recently created Pittsburgh Housing Authority, first spoke brjefly on the objectives of the Authority, the type of housing it expected to build, and the benefits this work would bring to Pittsburgh.

As the feature of the evening's entertainment, Mr. Evans exhibited some motion pictures of his recent trip through South and Central America. He gave an informal discussion during the showing of the film, laying particular emphasis on the masonry construction of the Incas and Mayans, who were the early inhabitants of these regions.

The meeting adjourned at 10:50 p. m., with 40 members and guests present.

University Heating Plant Described at Wisconsin Meeting

November 15, 1937. At a meeting of the Wisconsin Chapter, held at the City Club, Milwaukee, 27 were present for dinner and approximately 15 additional members and guests attended the technical session.

John Novotny, chief operating engineer, University of Wisconsin, was introduced by C. W. Miller. Mr. Novotny gave a history of the heating plant at the University, which first consisted of fireplaces and wood stoves in the North and South Halls, built in 1851 and 1855, respectively. The first power plant was installed in 1885 and was stationed in the old Mining and Metallurgy Laboratory, which is now a radio station. The second power plant was installed in 1895 and consisted of three 125-hp hand-fired boilers.

In 1906 the University consisted of a total of 23 buildings, while at the present date there are 110 buildings on the campus. Mr. Novotny told of the efficiency of the plant, which was raised from 60 per cent in 1918 to 74 per cent at the present time. All of the University buildings are heated by one central power plant having a full return of condensate from all buildings. The longest steam line is 6500 ft. Mr. Novotny expressed the opinion that with the addition of outside type temperature controls another 18 per cent of steam could be made available to heat the buildings, which burn up to 240 tons of coal per day. Lime and soda-ash method of water treatment is employed.

Other figures mentioned were a total of 627,000 sq ft of radiation, 3 miles of tunnel, 90,000 miles of piping, total yearly pumpage of water from Lake Mendota of 397,000,000 gal and total hot water gallons per year of 80,000,000. Both high and low pressure steam of 150 lb and 7 lb respectively are used. The plant produced 552,000,000 lb of steam yearly at a cost of 21¢ per 1000 lb of steam (coal only). The University grounds total 778 acres.

Secy. H. C. Frentzel reports that Mr. Novotny's talk was interesting and the figures regarding hot water and steam requirements, etc., were left on file for the information of anyone who wished to consider them further.

Noise Reduction Discussed at New York Chapter Meeting

January 17, 1938. Following dinner, the regular meeting of the New York Chapter was held at the Building Trades Club, New York City.

G. T. Stanton, manager, Technical Consulting Dept., Electrical Research Products, Inc., was the speaker for the evening and discussed the Problem of Noise in Ventilating and Methods of Approach to Its Solution.

Mr. Stanton, who acted as consultant in connection with acoustical problems in the design of the Lincoln Vehicular Tunnel in New York, defined noise in simple terms as the variation of pressure in the ear. He listed several methods of obtaining noise reduction among which are: (1) decreasing the resistance of vibrating parts; (2) reducing the forces which cause vibration; (3) reducing the area in vibration, and (4) reducing transmission of vibrating forces. In some types of electrical equipment it has been possible to set up sound waves in opposition to troublesome noises so that the two are out of phase and destroy each other. This, however, is not practical in mechanics.

According to the speaker, methods for reducing the resistance of vibrating parts, reducing the vibrating forces and reducing the area in vibration are all problems of the machine designer and usually must be taken care of before the apparatus reaches the field. In the field it is possible to reduce the vibrating forces that reach parts of the machine or system which are free to vibrate. One way of accomplishing this, for example, is to introduce springs between the vibrating force and the transmitting part.

The speaker cautioned his audience that in using acoustical material for sound absorption in ducts the duct must not be too large in cross-sectional area in relation to the wave length of the sound being transmitted. If the duct is large, the acoustical material should sub-divide the duct into a number of smaller sections, since sound tends to travel along the center line of the duct, and by breaking up the duct the transmitted sound is also broken up.

After a discussion of Mr. Stanton's interesting lecture the meeting was adjourned.

Dr. E. V. Hill Addresses Western Michigan Chapter

January 10, 1938. The regular meeting of the Western Michigan Chapter was held at the Mertens Hotel, Grand Rapids, with 63 present for dinner and 88 attending the meeting which followed.

Dr. E. Vernon Hill, Chicago, Ill., was the guest speaker and addressed the meeting on the subject of The Effect of Air Con-

ditions on Comfort, Health and Disease. Dr. Hill's investigations on human reactions to various conditions of humidity, heat, and air movement, aroused particular interest among the members. Using thermometers on nude subjects, it was determined that the lowest body temperatures occur in the hands and feet. Therefore, the first problem in air conditioning was precluded to be heating. It was Dr. Hill's conclusion that radiant heat is the key to air comfort. The comfort range for temperature established in Dr. Hill's test was 72 to 75F; for humidity 30 to 40 per cent; and for air movement, anything that was imperceptible.

In his general consideration of the subject, Dr. Hill described the advances in medical science due to the improvement in sanitary conditions, and noted that this had affected the death rate for intestinal diseases. It was then shown that death rates for lung diseases had remained unchanged over a similar period of time, and this fact was attributed by Dr. Hill to poor air. One of Dr. Hill's charts showed the death rate by months. Peak months were January, February and March; whereas, June, July and August were the correspondingly low months. Dr. Hill showed that in its medical aspects at least, it can be concluded from this that winter air conditioning is more important than summer conditioning. Dr. Hill stated that air conditioning would not be understood generally until the term *fresh air* was considered in terms of *quality*, expressing the opinion that there was no such thing as fresh air—that it was either clean, dirty, warm, cool, etc.

After Dr. Hill's talk, which was enjoyed by all, the meeting was adjourned.

Iowa-Nebraska Chapter Receives Charter from Pres. D. S. Boyden

December 14, 1937. The fourth meeting of the Iowa-Nebraska Chapter was held at 6:15 p. m. at the Hotel Kirkwood, Des Moines, Ia., with Vice-Pres. A. L. Walters presiding in the absence of Pres. M. J. Stevenson, and 19 members present.

The minutes of the last two meetings were read and approved. Following this Secy. W. R. White read a telegram from President Stevenson.

This was strictly a business meeting and the first order of business was the adoption of the rules and by-laws. Vice-President Walters was unanimously chosen as Chapter delegate to the 44th Annual Meeting of the A.S.H.V.E., and M. J. Stevenson was elected alternate delegate by ballot vote. A temporary Board of Governors was elected which included the present officers, as well as P. A. Norman, J. C. Knox and W. V. Hagan. It was moved and seconded that the appointment of the Nominating, Meetings and Publicity Committees be postponed until the January meeting.

Prof. P. A. Norman extended an invitation to the chapter to hold a meeting in conjunction with the Iowa State College Heating and Ventilating Conference to be held at Ames, Ia., sometime in the Spring.

E. H. Danielson moved that the meeting be adjourned, and L. L. Daubert seconded the motion.

November 19, 1937. The Iowa-Nebraska Chapter of the A.S.H.V.E. convened at its third session at the Fontenelle Hotel, Omaha, Nebr. Forty-three members and 6 guests attended.

In order to allow more time for the program planned for this meeting, it was moved that the reading of the minutes and all other business be deferred until the following meeting.

Col. D. S. Boyden, Boston, Mass., president of the A.S.H.V.E., presented a charter to the Iowa-Nebraska Chapter and following the presentation gave a brief resumé of the history and activities of the national organization.

President Boyden then addressed the chapter on the subject, Economic Use of Steam, and illustrated his talk with slides, showing the magnitude of the area served by the Boston Edison Co., the methods of construction, and arrangement of pipe lines.

The meeting adjourned at the conclusion of President Boyden's talk, and according to the report of Secy. W. R. White, proved to be an enjoyable and interesting meeting.

School Heating and Air Conditioning, Smoke Abatement Described at St. Louis Meetings

December 7, 1937. The regular meeting of the St. Louis Chapter was called to order at 7:45 p. m. by Pres. G. W. F. Myers at Garavelli's Restaurant. Twenty members and 15 guests were present.

The minutes of the previous meeting were read and approved, after which President Myers called for reports from the treasurer and the standing committees.

T. Skinker, president of the Engineers Club, then talked about plans for a joint meeting to be held the latter part of January at the Engineers Club.

E. A. Freund, who is in charge of air conditioning for the Union Electric Co., gave a few remarks regarding the progress made in air conditioning and what to expect in the coming year.

President Myers declared a five minute recess after which E. E. Carlson took the chair. He introduced John J. Carlisle, chief engineer of the Board of Education, who presented a paper on Heating and Air Conditioning the New Southwest High School. Mr. Carlisle gave a detailed report regarding all of the problems encountered in conditioning a large school. At the conclusion of his talk the speaker answered a number of questions which were of interest to all present.

After a rising vote of thanks to Mr. Carlisle the meeting was adjourned.

November 2, 1937. Sixteen members and 20 guests attended the regular meeting of the St. Louis Chapter which was called to order by Pres. G. W. F. Myers at Garavelli's Restaurant.

Secy. D. J. Fagin read the minutes of the previous meeting which were approved. C. E. Hartwein gave the treasurer's report.

E. E. Carlson introduced R. R. Tucker, commissioner of smoke abatement, who gave an interesting and constructive talk on the evils of smoke and the steps being taken to enforce the smoke ordinance.

At the conclusion of Mr. Tucker's discussion it was moved and seconded that the St. Louis Chapter go on record supporting the smoke abatement work and Secretary Fagin was instructed to write a letter to Mayor Dickmann assuring him that the local chapter would cooperate and assist in every way possible.

President Myers adjourned the meeting after a rising vote of thanks to Mr. Tucker.

Forced Air Heating Subject of Minnesota Meeting

January 10, 1938. Vice-Pres. J. E. Swenson called to order the meeting of Minnesota Chapter, held at the Men's Union, University of Minnesota, attended by 40 members and guests. Due to the absence of Secy. F. C. Winterer, M. H. Bjerken acted in his place and read a telegram from the New York Chapter, welcoming Minnesota members to the Society's Annual Meeting.

J. R. Hall, chairman of the Program Committee, was called upon to introduce the guest speaker, B. F. McLouth, chief engineer, Heater Division, Dail Steel Products Co., Lansing, Mich.

Starting with the history of forced air heating and the warm air industry, Mr. McLouth cited various codes that have been in effect and gave an outline of the new technical code. Mr. McLouth expressed the opinion that engineering facts, as he termed them, were of tremendous value to the industry in all fields, including that of sales.

F. W. Legler, Prof. F. B. Rowley, H. M. Betts and a number of guests presented questions to the speaker following his talk.

Massachusetts Chapter Discusses Hot Water Heating

January 18, 1938. Sixty-five members and guests of the Massachusetts Chapter braved the sub-zero New England weather to make the January meeting the largest of the season. A supper preceded the meeting held at Walker Memorial, Massachusetts Institute of Technology.

Pres. James Holt presided and presented the two speakers, A. W. Moulder, vice-president in charge of engineering, Grinnell Co., Providence, R. I., and Harold Alt, consulting engineer, New York, N. Y. Mr. Moulder presented an illustrated talk on Hot Water Heating Systems, covering especially recent developments in the field. Mr. Alt discussed in a clear manner the problems involved in writing these specifications and offered many suggestions to overcome difficulties often encountered.

According to the report of Secy. H. C. Moore, a general discussion of both topics followed the meeting.

Vice-President of Lamneck Products, Inc.

According to an announcement of P. S. Miller, president, Kimball Burr has joined Lamneck Products, Inc., Columbus, Ohio, as vice-president in charge of sales. Mr. Burr has been identified with the heating and air conditioning industry, having been associated with the B. F. Sturtevant Co., Inc., and the American Radiator Co. Mr. Burr is a member of the A.S.H.V.E. and is well known in the industry.

Members of I.H.V.E. Attend Annual Meeting and Exposition

Among those who attended the 44th Annual Meeting of the A.S.H.V.E. and the Fifth International Heating and Ventilating Exposition in New York were the following members of the *Institution of Heating and Ventilating Engineers* of Great Britain: W. M. Bailey, London, M. Sutcliffe, Sutcliffe Ventilating and Drying Co., Ltd., Manchester, H. J. Eley, Warwickshire, W. G. Case, Ideal Boilers & Radiators, Ltd., London, and L. J. Fowler, Richard Crittall Co., London.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 209 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by February 15, 1938, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
AARONSON, A. L., U. S. Govt., Philadelphia, Pa.	E. N. Black, 3rd H. H. Mather	L. P. Hynes W. C. Setzer (*Non-Member*)
ABBOTT, T. J., Vice Pres., George C. Abbott, Ltd., Toronto, Ont., Can.	J. H. Fox A. H. Hills	Thomas McDonald M. W. Shears
ADAM, R. W., Plbg. & Htg. Contractor, William A. Adam Co., Detroit, Mich.	L. L. McConachie W. G. Boales	F. J. Feely A. E. Knibb
ALFREY, H. F., Ch. Engr., Milwaukee Gas Specialty Co., Milwaukee, Wis.	C. H. Randolph E. A. Jones	Ernest Szekely J. H. Volk
ANDERSON, P. R., Oil Burner Installation Service, Radio Elec., Inc., Chester, Pa.	E. N. Black, 3rd H. H. Mather	W. F. Smith W. C. Setzer (*Non-Member*)
BASSETT, J. W., Sales Engr., McQuay, Inc., Detroit, Mich.	W. H. Old E. H. Clark	N. B. Hubbard W. C. Randall
BAY, C. H., Sales, Detroit Edison Co., Detroit, Mich.	S. S. Sanford G. D. Winans	G. H. Tuttle J. H. Walker
BENHAM, F. C., JR., Asst. Dist. Engr., Frigidaire Cond. Div., Fort Worth, Tex.	R. K. Werner H. W. Skinner	W. H. Badgett P. M. Bratten (*Non-Member*)
BEWS, JOHN, Dist. Sales Mgr., Can. Ice Mach. Co., Ltd., Montreal, Que., Can.	C. W. Johnson M. E. Dickenson	F. G. Phipps W. U. Hughes
BICKLEY, J. I., Engr., Nash-Kelvinator Corp., Detroit, Mich.	H. F. Hutzel F. J. Feely	J. A. Lee S. A. Whitt
BIGELOW, E. S., Mgr., Air Cond. Div., Trilling & Montague, Philadelphia, Pa.	H. H. Mather W. A. Bornemann	W. L. Yerkes M. E. Barnard
BITZER, R. D., Owner, R. D. Bitzer Co., Philadelphia, Pa.	C. W. Whitney R. C. Bolsinger	C. B. Eastman H. H. Mather
BOLAND, R. O., Mgr., Insulation Div., Alexander Murray & Co., Ltd., Montreal, Que., Can.	G. L. Wiggs L. H. Laffoley	W. U. Hughes A. F. Lamontagne

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
BOND, H. H., Partner, Edward E. Ashley, Cons. Engr., New York, N. Y.	R. H. Carpenter E. E. Ashley	C. L. Chase H. W. Fiedler
BORG, E. H., Member of Firm, Proudfoot-Rawson-Brooks & Borg, Des Moines, Iowa.	T. R. Johnson L. L. Daubert	B. F. Elbert G. S. Stiles
BOTTUM, E. W., Mech. Engr., Nash-Kelvinator Corp., Detroit, Mich.	A. E. Knapp H. F. Hutzel	J. A. Lee H. C. Van Nouhuys
BRADLEY, J. M., Br. Mgr., Airtherm Mfg. Co., Chicago, Ill.	R. J. Tenkonohy L. W. Moon	G. W. F. Myers R. C. Driemeyer
BRASHAW, C. J., Installation Mgr., J. F. Stampfer Co., Dubuque, Iowa.	A. R. MacMillan F. H. Williams	J. S. Brown, Jr. Albert Buenger
BRITTAIN, ALFRED, JR., Engr., Weathermakers (Can.) Ltd., Toronto, Ont., Can.	W. C. Oke G. A. Playfair	M. L. Woollard H. H. Angus
BROWN, M. W., Sales Engr., American Blower Corp., Dallas, Texas.	C. L. Kribs, Jr. L. L. Landauer	O. H. Mehl G. H. Meffert
BUCKERIDGE, V. L., Partner, H. Buckeridge & Son, Grosse Pointe Park, Mich.	L. L. McConachie W. G. Boales	A. E. Knibb F. J. Feely
BULLER, C. R., Asst. Ch. Engr., Heil Co., Milwaukee, Wis.	H. C. Frentzel C. H. Randolph	Ernest Szekely W. A. Ouweneel
CAMPBELL, BOWEN, Engr., Campbell Htg. Co., Des Moines, Iowa.	T. R. Johnson L. L. Daubert	L. E. Hedeen B. F. Elbert
CARDWELL, M. R., Htg. Engr., Haller Appliance Sales Co., St. Louis, Mo.	A. L. Haller E. E. Carlson	G. W. F. Myers D. J. Fagin
CARROLL, A. F., Development Engr., Automatic Prod. Co., Milwaukee, Wis.	S. G. Swisher, Jr. L. W. Hessler	J. H. Volk C. H. Randolph
CARROLL, W. M., Engr., Tom Dolan Htg. Co., Oklahoma City, Okla.	R. G. Dolan E. F. Dawson	E. W. Gray S. L. Rolland
CHALMERS, C. H., Owner & Gen. Mgr., Chalmers Oil Burner Co., Minneapolis, Minn. (*Reinstatement*)	F. B. Rowley A. B. Algren	J. V. Martenis H. E. Gerrish
CHASE, L. R., Dist. Supervisor, The Carters-Waters Corp., Kansas City, Mo. (*Advancement*)	C. A. Weiss N. W. Downes	D. D. Zink A. L. Maillard
CHENEVERT, J. G., Consulting Engr., Arthur Surveyer & Co., Montreal, Que., Can.	Henri Gendron G. L. Wiggs	C. W. Johnson W. W. Timmins
CHRONE, R. E., Design Engr., R. F. Taylor, Houston, Texas.	R. F. Taylor J. A. Kiesling	J. D. Morrow R. J. Salinger
CITRON, D. J., Student, Kansas University, Lawrence, Kan.	A. H. Sluss Gustav Nottberg	Henry Nottberg, Jr. A. L. Maillard
CLAR, ROBERT, JR., Br. Mgr., U. S. Radiator Corp., Baltimore, Md.	R. F. Connell F. H. Blackmore	W. A. Russell R. K. Milward
CLARKSON, J. R., Student, David Ranken Jr. School of Mech. Trades, St. Louis, Mo.	G. B. Rodenheiser E. E. Carlson	G. W. F. Myers C. E. Hartwein
CONROY, M. J., Dist. Mgr., Reynolds Corp., Upper Darby, Pa.	H. H. Mather C. H. Schneider	C. B. Eastman W. F. Smith
COOK, A. L., Engr., A. & M. College of Texas, College Station, Texas.	W. E. Long F. E. Giesecke	G. M. Hines W. H. Badgett
COOK, H. D., Control Engr., R. L. Deppmann Co., Detroit, Mich.	R. L. Deppmann L. A. Burch	W. G. Boales F. J. Feely
COOPER, D. S., Air Cond. Engr., Air Cond. Co., Houston, Texas. (*Advancement*)	I. E. Rowe J. A. Walsh	W. H. Badgett F. E. Giesecke
CRAMER, W. G., Sales Mgr., The Marley Co., Cincinnati, Ohio.	K. B. Little W. H. Junker	C. J. Kiefer L. G. Powers
CROUT, M. M., Br. Mgr., York Ice Mach. Corp., Atlanta, Ga.	L. J. Du Bois J. R. Hertzler	I. C. Baker G. E. Tuckerman
CUMNOCK, H., Secy. & Treas., Little Rock Refrigeration Co., Inc., Little Rock, Ark.	F. H. Williams A. R. MacMillan	Albert Buenger Herman Worsham
DAVIS, G. L., Jr., Estimator, L. L. McConachie Co., Detroit, Mich.	L. L. McConachie A. E. Knibb	W. G. Boales E. H. Clark
DELL'ORTO, LUCIANO, Gen. Engr., Office of Giuseppe Dell'Orto, Milano, Italy.	Aldo Gini E. A. Russell	A. V. Hutchinson C. C. Elmes (*Non-Member*)
DEMPSEY, S. J., Plbg., Htg. & Vtg., Stephen J. Dempsey Co., Battle Creek, Mich.	S. W. Todd, Jr. T. L. Osberger	T. D. Stafford H. D. Bratt
DISTEL, R. E., Engr., Distel Htg. Equip. Co., Lansing, Mich.	W. W. Bradfield S. W. Todd, Jr.	B. F. McLouth D. L. Taze
DOYLE, B. J., Sales Engr., Weil-McLain Co., Detroit, Mich.	L. L. McConachie W. G. Boales	A. E. Knibb F. J. Feely
DREHER, L. F., Student, David Ranken Jr. School of Mech. Trades, St. Louis, Mo.	G. B. Rodenheiser C. E. Hartwein	D. J. Fagin G. W. F. Myers
DUPLANT, J. L., Air Cond. Specialist, Westinghouse Elec. Internatl. Co., New York, N. Y.	R. C. Bonthron H. W. Fiedler	B. R. Massey (*Non-Member*) J. A. Heller (*Non-Member*)
ECKART, J. H., Adjuster, Sears, Roebuck & Co., Philadelphia, Pa.	E. N. Black, 3rd L. P. Hynes	H. H. Mather W. C. Setzer (*Non-Member*)
ELLIOT, G. B., Sales Engr., Francis Hankin Co., Montreal, Que., Can.	G. L. Wiggs W. W. Timmins	C. W. Johnson R. R. Noyes
ELLIS, L. W., Sales Engr., Disney-Leffel Co., Oklahoma City, Okla.	A. A. Hoppe Dean Spencer	S. L. Rolland E. W. Gray
EMERSON, J. J., Sales Engr., Trane Co. of Can., Ltd., Montreal, Que., Can.	G. L. Wiggs J. R. Dupuis	W. W. Timmins C. W. Johnson

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
EUTSLER, E. E., Jr., Engr., Buffalo Forge Co., Buffalo, N. Y.	W. R. Heath Roswell Farnham	A. A. Criqui R. D. Madison
FAMILETTI, A. R., Ch. Engrg. Draftsman, Navy Yard, Philadelphia, Pa. (*Advancement.*)	F. D. Mensing E. N. Sanbern	L. C. Davidson M. F. Blankin
FELLOWS, J. R., Assoc. in Mech. Engrg., University of Illinois, Urbana, Ill.	W. H. Severns E. L. Broderick	M. K. Fahnestock P. O. Mohn (*A.S.M.E.*)
FERDERBER, M. B., In Charge of Fever Therapy Research Fund, University of Pittsburgh, Pittsburgh, Pa.	F. C. Houghten M. L. Carr	C. M. Humphreys T. F. Rockwell
FERGESTAD, M. L., Field Engr., Bark Prod. Div., Pacific Lumber Co., Minneapolis, Minn., (*Advancement*).	R. E. Backstrom H. M. Betts	F. B. Rowley A. B. Algren
FINK, W. N., Sales Engr., John Landers, Mfrs. Agent, Buffalo, N. Y.	Joseph Davis J. J. Landers	B. C. Candee S. R. Rente (*Non-Member*)
FLEAK, W. D., Lab. Instructor, Refrigeration & Air Cond. Inst., Chicago, Ill.	E. V. Hill S. S. Sommerfield	F. L. Howard J. J. Aeberly
FLUM, J. C., Metal Polisher, Henry Disston & Son, Philadelphia, Pa.	E. N. Black, 3rd H. H. Mather	L. P. Hynes W. C. Setzer (*Non-Member*)
FOSTER, E. W., Adjuster in Plbg. & Htg., Sears, Roebuck & Co., Philadelphia, Pa.	E. N. Black, 3rd L. P. Hynes	H. H. Mather W. C. Setzer (*Non-Member*)
FRANKLIN, S. D., Engr., Standard Oil Co. of Pa., Philadelphia, Pa.	E. N. Black, 3rd H. H. Mather	L. P. Hynes W. C. Setzer (*Non-Member*)
FRANKLIN, S. H., Jr., Htg. Contractor, S. H. Franklin, Jr., Lynchburg, Va.	A. R. MacMillan F. H. Williams	W. A. Chapman, Jr. Herman Worsham
GALLAGHER, F. H., Asst. Ch. Engr., Bd. of Public Education, Pittsburgh, Pa.	G. P. Ellis F. C. McIntosh	J. F. Collins, Jr. G. S. McEllroy (*Non-Member*)
GARBER, W. E., Jr., Field-Engr., Farquhar Furnace Co., Cincinnati, Ohio.	C. L. Sapp V. G. Marconett	H. E. Sproull J. T. Fraig (*A.S.M.E.*)
GERSTENBERGER, E. J., Sales Engr., F. R. Dengel Co., Milwaukee, Wis.	J. H. Volk C. H. Randolph	T. M. Hughey C. W. Miller
GIFFORD, E. W., Chief Engr., Airtemp Construction Corp., Detroit, Mich. (*Reinstatement & Advancement*)	H. E. Paetz G. H. Cummins	F. R. Dickenson E. L. Hogan
GILMORE, J. L., Owner, John L. Gilmore, Brunswick, Ga.	C. M. McGarvey (*Non-Member*) L. S. Mieur (*Non-Member*)	J. L. Robeson (*Non-Member*) O. W. Kothe (*A.S.M.E.*)
GRAY, J. W., Member of Firm, The Gray-Henry Co., Bay City, Mich.	E. B. Root F. R. Bishop	Jay Barton B. F. McLouth
GROSS, J. A., Gen. Mgr., Wynne Construction Service, Philadelphia, Pa.	E. N. Black, 3rd H. H. Mather	W. F. Smith W. C. Setzer (*Non-Member*)
HAERLE, R. A., Mech. Engr., Bayley Blower Co., West Allis, Wis.	Ernest Szekely C. H. Randolph	J. H. Volk T. M. Hughey
HALEY, R. T., Supervisor House Htg. Dept., Minneapolis Gas Light Co., Minneapolis, Minn.	H. M. Betts H. C. Mills	A. B. Algren J. C. Davidson
HALL, W. L., Engr., Frank A. Leon Co., Washington, D. C.	P. H. Loughran, Jr. E. V. Fineran	W. H. Loving L. S. Ourusoff
HALLSTEIN, H. T., Sales Engr., Delco-Frigidaire Cond. Div., Dayton, Ohio.	Herman Worsham Albert Buenger	F. H. Williams C. E. Lewis
HAMACHER, K. F., Partner, Hamacher & Williams, Milwaukee, Wis.	S. G. Swisher, Jr. J. H. Volk	C. H. Randolph L. W. Hessler
HAMLET, T. F., Sales Engr., Taylor Forbes, Ltd., Montreal, Que., Can.	F. A. Hamlet Leo Garneau	F. G. Phipps A. B. Darling
HARRISON, J. C., Design Engr., Texas Air Cond. Co., Fort Worth, Texas.	W. H. Badgett F. E. Giesecke	G. M. Hines W. E. Long
HAUPT, H. F., Htg. Sales, Kohler Co., Kohler, Wis.	J. H. Volk C. H. Randolph	S. G. Swisher, Jr. Ernest Szekely
HAWISHER, H. H., Engr., Automatic Htg. & Engrg. Co., Lima, Ohio.	A. R. MacMillan F. H. Williams	Albert Buenger J. S. Brown, Jr.
HAZLETT, T. L., Medical Director, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa.	F. C. Houghten M. L. Carr	C. M. Humphreys T. F. Rockwell
HEDEEN, L. F., Sales Engr., The Trane Co., Des Moines, Iowa.	T. R. Johnson M. L. Todd	J. C. Knox W. R. White
HENRY, E. C., Member of Firm, The Gray-Henry Co., Bay City, Mich.	F. R. Bishop E. B. Root	Jay Barton B. F. McLouth
HICKMAN, H. V., Sales Engr., Birchfield Boiler Co., San Francisco, Calif.	W. W. Cockins G. M. Simonson	G. J. Cummings C. E. Bentley
HILL, V. H., Engr., Nash-Kelvinator Corp., Detroit, Mich.	A. E. Knapp H. F. Hutzel	S. A. Whitt J. A. Lee
HIRSCH, M. H., Sales Engr., Hoffman Specialty Co., Inc., New York, N. Y.	E. N. Sanbern J. K. Peacock	A. C. Roy Potter Bowles
HOFFMAN, ANGELO, Vice Pres., Louis Hoffman Co., Milwaukee, Wis.	C. W. Miller T. M. Hughey	J. H. Volk J. G. Hammerschlag (*A.S.R.E.*)
HOLT, W. H., Engr., Buffalo Forge Co., Buffalo, N. Y.	A. A. Criqui W. R. Heath	R. D. Madison Roswell Farnham
HOLUBA, H. J., Sales Engr., A. J. Mfg. Co., Kansas City, Mo.	C. A. Weiss M. A. Disney	W. L. Cassell H. S. White
HUGHES, H. R., Mgr., Fuel Oil Div., McColl-Frontenac Oil Co., Ltd., Montreal, Que., Can.	G. L. Wiggs W. U. Hughes	W. W. Timmins L. H. Laffoley

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
HUMPHREY, L. G., Jr., Engr., Buffalo Forge Co., Buffalo, N. Y.	W. R. Heath A. A. Criqui	R. D. Madison Roswell Farnham
HUNTER, H. A., Secy.-Treas. & Engr., Texas Air Cond. Corp., Fort Worth, Tex.	H. W. Skinner R. K. Werner	J. B. Hawley (*A.S.C.E.*) M. C. Nichols (*A.S.C.E.*)
JARDINE, W. H., JR., Pres., Iona Ventilator Co., Inc., Philadelphia, Pa.	E. N. Black, 3rd L. P. Hynes	H. H. Mather W. C. Setzer (*Non-Member*)
JOHNSON, T. R., Br. Mgr., The Trane Co., Des Moines, Iowa. (*Reinstatement*)	M. L. Todd A. L. Walters	W. R. White R. A. Norman
JOHNSTON, R. M., Steam & Gas Htg. Sales, N. Y. State Elec. & Gas Corp., Lockport, N. Y.	L. P. Saunders B. C. Candee	M. C. Beman P. S. Hedley
JUNGBLUTH, E. N., Sales Engr., Linde Can. Refrigeration Co., Montreal, Que., Can.	G. L. Wiggs C. W. Johnson	F. G. Phipps G. H. Osborne
KAISER, C. W., Installation Engr., Powers Regulator Co., New York, N. Y.	P. B. Gordon W. E. Heibel	John Everetts, Jr. John James
KEITH, J. P., Engr., Canadian Dominion Engrg. Co., Montreal, Que., Can.	F. G. Phipps W. W. Timmins	C. W. Johnson G. L. Wiggs
KELLY, J. F., Vice Pres. & Ch. Engr., Kelly Sales Corp., Arlington, Mass.	G. B. Gerrish H. L. Farrow	C. G. Lauckner, 3rd R. A. Sheffield
KESSLER, C. F., Asst. Prof. Mech. Engrg., University of Mich., Ann Arbor, Mich.	J. E. Emswiler Axel Marin	H. E. Keeler (*A.S.M.E.*) F. A. Mickle (*A.G.M.A.*)
KILLOUGH, R. E., Service Dept., Standard Oil Co. of Pa., Philadelphia, Pa.	E. N. Black, 3rd H. H. Mather	W. F. Smith W. C. Setzer (*Non-Member*)
KINDORF, ORLAN, Vice Pres. & Mgr., General Air Cond. Co., Sacramento, Calif.	James Gayner C. E. Bentley	G. M. Simonson C. S. Ames
KLUGE, B. M., Sales Engr., Bayley Blower Co., Milwaukee, Wis.	Ernest Szekely C. H. Randolph	W. A. Ouweneel R. G. Koch
KNEPPER, H. H., Erection Supt., Minneapolis-Honeywell Reg. Co., Omaha, Nebr.	Paul Larkin F. L. Banner	W. R. White M. B. Peiser
KOLB, F. W., Factory Repr., American Air Filter Co., San Francisco, Calif.	C. E. Bentley G. M. Simonson	James Gayner G. J. Cummings
KOREN, N. A., Design Engr., Minneapolis, Minn.	F. W. Legler P. S. Podany (*Non-Member*)	H. B. Williams (*Non-Member*) H. E. Kreher (*Non-Member*)
KRATZ, R. W., Dist. Engr., Delco-Frigidaire Cond. Div., Los Angeles, Calif. (*Reinstatement*)	E. H. Kendall W. D. Fabling	H. B. Lauer Harrold English
KUGEL, H. K., Smoke Regulation Engr., Govt. of D. C., Washington, D. C.	T. H. Urdahl H. M. Nobis	A. L. Vanderhoof J. P. Jones
KUREK, T. C., Mech. Engr., Henrici-Lowry Engrg. Co., Kansas City, Mo.	D. D. Zink Henry Nottberg	Sidney Pines Emil Haas, Jr.
LAFONTAINE, E. A., Sales Mgr., Kelvinator of Can., Ltd., Montreal, Que., Can.	Leo Garneau F. A. Hamlet	F. G. Phipps A. B. Darling
LANDES, B. F., Consulting & Design Engr., Des Moines, Iowa.	A. L. Walters T. R. Johnson	L. E. Hedeen L. L. Daubert
LANDES, J. M., Wholesale Mgr., Controlled Air Corp., St. Louis, Mo.	L. R. Szombathy V. E. Hugoniot	L. J. Sydow L. L. Hamig
LASETER, F. L., Mgr. Central Heat Dept., Atlanta Gas Light Co., Atlanta, Ga.	T. T. Tucker L. F. Kent	W. P. Sherman C. T. Baker
LAWRENCE, F. D., Sales Repr., Clarage Fan Co., New York, N. Y.	C. S. Pabst W. E. Heibel	W. T. Wrightson R. H. Wadsworth
LEEK, C. W., Ch. Engr. & Mgr., Leek & Co., Ltd., Vancouver, B. C., Can.	Walter Leek S. D. Peterson	H. J. McCreery E. H. Gurney
LICANDRO, J. P., Engr., Carrier Corp., Syracuse, N. Y.	Lawrence Macrow R. E. Cherne	L. L. Lewis W. A. Grant
LINGEN, R. A., Engr., American Fdry. & Furnace Co., Milwaukee, Wis.	C. H. Randolph J. H. Volk	T. M. Hughey E. A. Jones
LOCKHART, C. W., Engr., Buffalo Forge Co., Buffalo, N. Y.	W. R. Heath A. A. Criqui	R. D. Madison Roswell Farnham
LONG, H. P., Sales Engr., Buffalo Forge Co., Buffalo, N. Y.	A. A. Criqui W. E. Voisinet	Roswell Farnham W. R. Heath
LO PICCOLO, A. A., Contract Engr., Carrier Corp., Syracuse, N. Y.	Lawrence Macrow R. E. Cherne	W. A. Grant W. B. Rorison (*A.S.R.E.*)
LOWE, R. A., Engr., Buffalo Forge Co., Buffalo, N. Y.	W. R. Heath A. A. Criqui	R. D. Madison Roswell Farnham
LOWE, WALTER, Service Man, Peoples Natural Gas Co., Pittsburgh, Pa.	C. M. Humphreys T. F. Rockwell	R. B. Stanger P. A. Edwards
LYKE, H. W., Ch. Engr., Peter Smith Heater Co., Detroit, Mich.	R. L. Deppmann L. A. Burch	G. D. Winans G. H. Tuttle
LYNN, R. G., Htg. Engr., Frigidaire Div., St. Paul, Minn.	H. E. Gerrish H. S. Morton	F. B. Rowley H. M. Betts
MACEACHIN, G. C., Dist. Engr., Frigidaire Div., Fort Worth, Texas.	R. K. Werner H. W. Skinner	W. H. Badgett P. M. Bratten (*Non-Member*)
MACK, E. H., Asst. Sales Mgr., Vilter Mfg. Co., Milwaukee, Wis.	C. H. Randolph Ernest Szekely	E. A. Jones J. H. Volk
MACLACHLAN, V. D., Sales Engr., Minneapolis-Honeywell Reg. Co., Montreal, Que., Can.	C. W. Johnson R. R. Noyes	A. W. Peart F. R. Barnsley
MAGEE, K. B., Air Cond. Engr., P. R. Moses & Associates, New York, N. Y.	W. E. Heibel John Everetts, Jr.	C. H. Taze F. Z. Goosman (*Non-Member*)

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
MAHON, C. A., Pres. & Mgr., Air Control Equip. Co., Kansas City, Mo.	David Caleb M. A. Disney	A. L. Maillard A. D. Marston
MARCHIO, EMILIO, JR., Student, Kansas University, Lawrence, Kan.	A. H. Sluss Gustav Nottberg	Henry Nottberg, Jr. A. L. Maillard
MARZORATI, GIUSEPPE, Consulting Engr., Societa Giacomo Tucker, Milano, Italy.	E. J. Ritchie R. W. Cumming	T. N. Adlam Luciano Dell'Orto
MAY, A. O., Sales Engr., Stannard Power Equip. Co., Chicago, Ill. (*Reinstatement & Advancement*)	J. M. Stannard J. J. Hayes	Tom Brown C. F. Hayden
MAYER, R. L., Sales Engr., Newton Supply Co., Lansdowne, Pa.	E. N. Black, 3rd H. H. Mather	L. P. Hynes W. C. Setzer (*Non-Member*)
McCAFFRAY, C. E., Design Engr., Henry Adams, Inc., Baltimore, Md.	E. L. Crosby J. deB. Shepard	J. E. Seiter F. T. Leilich (*A.S.M.E.*)
McDONALD, J. J., Research Engr., Nash-Kelvinator Corp., Detroit, Mich.	H. C. Van Nouhuys S. A. Whitt	A. E. Knapp G. D. Winans
McKERLIE, JARDINE, Editor, Refrigeration & Air Cond., Gardenvale, Que., Can.	C. W. Johnson A. B. Darling	G. L. Wiggs F. G. Phipps
McREYNOLDS, C. V., Engr., Natkin & Co., Des Moines, Iowa.	Henry Kleinkauf W. R. White	A. L. Walters R. A. Norman
MILLER, D. C., Office of Alfred Kellogg, Boston, Mass.	Alfred Kellogg H. B. Wiegner	W. T. Jones N. J. Shaw
MILNE, A. H., Dir. Dept. of Bldgs., City of Montreal, Montreal, Que., Can.	G. L. Wiggs W. W. Timmins	J. R. Dupuis M. F. Dixon
MIRABILE, J. J., Field Service Engr., Elliott-Lewis Co., Philadelphia, Pa.	E. N. Black, 3rd H. H. Mather	L. P. Hynes W. C. Setzer (*Non-Member*)
MOFFAT, O. G., Application Engr., Can. Westinghouse Co., Ltd., Hamilton, Ont., Can. (*Advancement*)	H. R. Roth M. W. Shears	Ernest Fox H. D. Henion
MOLFINO, PHILIP, Ch. Draftsman, Leland & Haley, San Francisco, Calif.	H. S. Haley W. E. Leland	C. E. Bentley G. M. Simonson
MONTGOMERY, E. G., Special Repr., Steel Co. of Can., Montreal, Que., Can.	G. L. Wiggs T. H. Worthington	F. J. Friedman J. A. M. Robertson
MORIN, A. R., Mgr. Refrigeration Dept., Macklanburg Brass & Copper, Inc., Oklahoma City, Okla.	Dean Spencer A. A. Hoppe	S. L. Rolland E. W. Gray
MORRISON, W. L., Gen. Mgr., Fair Plbg. & Htg. Co., Great Bend, Kan.	W. F. Ryan C. W. Adams	A. H. Sluss E. W. Seaman (*Non-Member*)
MUESSIG, J. W., Sales Engr., Clarage Fan Co., Chicago, Ill.	D. B. Scheidecker C. R. McConner	S. H. Downs H. A. Brinker
MUNKELT, F. H., Vice Pres., W. B. Connor Engr. Corp., New York, N. Y.	H. C. Murphy John Hellstrom	A. F. Hinrichsen H. D. Edwards (*A.S.R.E.*)
NATHAN, P. V., Ch. Draftsman, Linde Can. Refrigeration Co., Ltd., Montreal, Que., Can.	C. W. Johnson G. L. Wiggs	F. G. Phipps R. R. Noyes
NELSON, R. O., Htg. & Air Cond. Engr., Fedders Mfg. Co., Chicago, Ill.	E. M. Mittendorff C. E. Murray	J. C. Hines W. A. Kuechenberg
NININGER, C. H., Dist. Repr., H. C. Baker Co., Inc., Roanoke, Va.	A. R. MacMillan F. H. Williams	W. A. Chapman, Jr. J. J. La Salvia
NORRINGTON, W. L., Engr., Minneapolis-Honeywell Reg. Co., Cleveland, Ohio.	R. A. Wilson Philip Cohen	H. E. Wetzell C. F. Eveleth
OBERSCHULTE, R. H., Sales Engr., D. T. Randall & Co., Detroit, Mich.	R. D. Randall E. H. Clark	J. S. Kilner W. G. Boales
OLCHOFF, MAURICE, Br. Mgr., Natkin & Co., Des Moines, Iowa. (*Reinstatement*)	Henry Kleinkauf W. R. White	A. L. Walters R. A. Norman
OLIVIERI, P. D., Plbg. & Htg., Herman Wahl, Philadelphia, Pa.	E. N. Black, 3rd H. H. Mather	L. P. Hynes W. C. Setzer (*Non-Member*)
PASSUR, N. A., Air Cond. Engr., Southern Pacific Co., San Francisco, Calif.	W. W. Cockins N. H. Peterson	C. E. Bentley G. J. Cummings
PAWKETT, L. S., Mfrs. Repr., L. S. Pawkett, San Antonio, Texas.	M. L. Diver L. P. Halleck	R. W. Kotzebue A. J. Rummel
PETERSON, C. L., Br. Mgr., Minneapolis-Honeywell Reg. Co., San Francisco, Calif.	C. E. Bentley G. M. Simonson	James Gayner L. H. Cochran
PLAAG, A. F., Owner, Frezan Refrigeration Service, Trenton, N. J.	E. N. Black, 3rd H. H. Mather	L. P. Hynes W. C. Setzer (*Non-Member*)
PODOLSKE, A. R., Prop., Sheet Metal Works, Milwaukee, Wis.	W. A. Ouweneel C. H. Randolph	Ernest Szekely J. H. Volk
PORTER, N. E., Engr., General Air Cond. Co., Sacramento, Calif.	G. M. Simonson C. E. Bentley	James Gayner C. S. Ames
PROEBSTLE, LEONARD, Air Cond. Engr., Frigidaire Div., Minneapolis, Minn.	F. B. Rowley A. B. Algren	C. E. Lund R. C. Jordan
REILLY, B. B., Engr., Trilling & Montague, Philadelphia, Pa.	H. H. Mather W. A. Bornemann	W. L. Yerkes M. E. Barnard
RHINE, G. R., Air Cond. Sales Engr., San Antonio Public Service Co., San Antonio, Tex.	A. J. Rummel L. O. Vogelsang (*A.S.M.E.*)	R. E. Sivley (*A.S.M.E.*) F. A. Covington (*I.E.S.*)
RUSSELL, T. W., Sales Engr., L. J. Mueller Furnace Co., Minneapolis, Minn.	W. C. Kuehn H. M. Betts	J. C. Davidson J. E. Swenson
RYBOLT, A. L., Asst. Gen. Mgr., Rybolt Heater Co., Ashland, Ohio.	J. H. Van Alsburg G. F. Nightingale	A. W. Williams F. W. Wills
SAMPSON, E. T., Mgr., Acoustical Dept., Atlas Asbestos Co., Ltd., Montreal, Que., Can.	G. L. Wiggs R. R. Noyes	C. W. Johnson W. W. Timmins

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
SCHLICHTER, C. F., Dist. Mgr., Surface Combustion Corp., Washington, D. C.	P. B. Holmes A. E. Stack	E. V. Fineran E. H. Lloyd
SCHMID, J. U., Sales Engr., Louis Allis Co., Milwaukee, Wis.	W. A. Ouweneel T. M. Hughey	J. H. Volk S. G. Swisher, Jr.
SCHMIDT, E. G., Consulting Engr., Berlin-Friedenau, Germany.	K. Brand (*VDJ*) J. Mautner (*VDJ*)	W. F. Lindner (*VDJ*) Dipl. Ing. Yungerhaus (*VDJ*)
SCHMIELER, J. B., Research Engr., A.S.H.V.E. Research Lab., Pittsburgh, Pa.	F. C. Houghten M. L. Carr	C. M. Humphreys T. F. Rockwell
SCHNELL, R. H., Asst. Mech. Engr., B. E. Landes, Des Moines, Iowa.	A. L. Walters T. R. Johnson	L. E. Hedeen L. L. Daubert
SCHOENHOFEN, L. H., Jr., Design Engrg. Dept., Okla. Gas & Elec. Co., Oklahoma City, Okla.	S. L. Rolland J. H. Carnahan	E. W. Gray R. G. Dolan
SCHULENBERG, F. W., Jr., Air Cond. Sales Engr., Frigidaire Div., St. Louis, Mo.	A. R. MacMillan F. H. Williams	G. W. F. Myers H. M. Strange (*Non-Member*)
SEARS, C. B., Design Engr., Buffalo Forge Co., Buffalo, N. Y.	A. A. Criqui Roswell Farnham	R. D. Madison W. R. Heath
SHORE, DAVID, JR. Research Engr., A.S.H.V.E. Research Lab., Pittsburgh, Pa.	F. C. Houghten M. L. Carr	C. M. Humphreys T. F. Rockwell
SITTERLY, C. T., In charge of Installation, Andrew J. Bachofer, Salina, Kan.	W. F. Ryan C. W. Adams	H. J. Nottberg H. J. Nottberg, Jr.
SKELLEY, J. H., Educational Instructor, Delco-Frigidaire Sales Corp., Dayton, Ohio.	A. R. MacMillan F. H. Williams	C. E. Lewis Albert Buenger
SPURNEY, F. H., Bldg. Mgr., Federal Reserve Bldg., Washington, D. C.	S. L. Gregg S. P. Eagleton	L. S. Ourusoff E. H. Lloyd
SPARK, WILLIAM, Mgr., Insulation Dept., Atlas Asbestos Co., Ltd., Montreal, Que., Can.	G. L. Wiggs R. R. Noyes	C. W. Johnson W. W. Timmins
STAFFORD, J. F., Sales, McQuay, Inc., Minneapolis, Minn.	C. E. Gausman E. H. Seelert	C. L. Bensen W. T. Jones
STARR, LYNDON, Field Repr., Refrigeration & Air Cond. Inst., University City, Mo.	C. F. Boester J. W. Cooper	G. W. F. Myers D. J. Fagin
STASZESKY, F. M., Apprentice Plbg. & Htg., Longwood Farms, Longwood, Pa.	F. I. Ponsell H. E. Grossman	R. W. Taylor (*Non-Member*) C. H. Brown (*Non-Member*)
STEINER, T. J., Sales Engr., Pomona Sheet Metal Works, Lawndale, Calif.	E. H. Kendall H. B. Lauer	J. F. Park Harrold English
STENCEL, R. A., Ch. Engr., Can. Ice Mach. Co., Ltd., Toronto, Ont., Can.	H. R. Roth M. W. Shears	Thomas McDonald J. H. Fox
STEVENS, A. L., Refrigerating Engr., Mollenberg-Betz Mach. Co., Buffalo, N. Y.	H. J. Mollenberg W. R. Heath	A. A. Criqui Roswell Farnham
SULLIVAN, C. J., Owner & Mgr., C. J. Sullivan, Philadelphia. Pa.	E. N. Black, 3rd H. H. Mather	L. P. Hynes W. C. Setzer (*Non-Member*)
TAYLOR, FIELDING, JR., Sales Engr., Buffalo Forge Co., Buffalo, N. Y.	A. A. Criqui Roswell Farnham	W. R. Heath R. D. Madison
TAYLOR, R. B., Sales Engr., Buffalo Forge Co., Buffalo, N. Y.	C. C. Cheyney W. R. Heath	Roswell Farnham A. A. Criqui
TERRILL, MARK, Sales Repr., Ill. Philip Carey Co., Grand Rapids, Mich.	T. D. Stafford W. W. Bradfield	C. H. Morton T. L. Osberger
THOMPSON, E. B., Htg. Engr., Cincinnati Gas & Elec. Co., Cincinnati, Ohio.	C. E. Hust R. W. Sigmund	H. E. Sproull F. J. Ward
THULMAN, R. K., Mech. Engr., Federal Housing Administration, Washington, D. C.	C. E. Mayette W. F. Goddard	T. H. Urdahl L. F. Nordine
TOWLE, P. H., Engr., General Air Cond. Co., Sacramento, Calif.	G. M. Simonson C. E. Bentley	James Gayner C. S. Ames
TUPPER, G. B., Sales Mgr., General Regulator Corp., Chicago, Ill. (*Reinstatement*)	E. M. Mittendorff C. W. DeLand	C. W. Johnson S. I. Rottmayer
VARNEY, F. H., Jr., Engr. Air Cond. Dept., General Elec. Co., San Francisco, Calif.	G. M. Simonson C. E. Bentley	C. S. Ames James Gayner
VOLLMANN, C. W., Pres., Linde Can. Refrigeration Co., Ltd., Montreal, Que., Can.	W. U. Hughes C. W. Johnson	G. L. Wiggs F. J. Friedman
WAGNER, E. K., Sales Engr., Powers Regulator Co., Philadelphia, Pa.	L. P. Hynes W. F. Smith	H. H. Mather R. J. Cowell (*Non-Member*)
WASSON, R. A., Eastern Dist. Mgr., Clarage Fan Co., New York, N. Y.	C. S. Pabst John Everetts, Jr.	R. H. Wadsworth W. E. Heibel
WEID, H. L., Test Engr., Hynes Elec. Htg. Co., Philadelphia, Pa.	L. P. Hynes J. T. Barron	H. H. Mather W. F. Smith
WEIL, F. H. E., Asst. Ch. Engr., Standard Distributing Corp., Milwaukee, Wis.	C. H. Randolph W. A. Ouweneel	W. F. Noll J. H. Volk
WELLS, E. P., Air Cond. Engr., Gay Engrg. Corp., Los Angeles, Calif.	L. L. McConachie W. G. Boales	A. E. Knibb G. D. Winans
WHITE, R. C., Sales Engr., Carrier Corp., Chicago, Ill.	E. P. Heckel E. T. Murphy	E. M. Mittendorff Henry Lauterbach, Jr.
WHITESELL, R. H., Sales Repr., Bryant Air Cond. Corp., Minneapolis, Minn.	H. M. Betts H. C. Mills	A. B. Algren J. C. Davidson
WILDE, R. S. M., Consulting Mech. Engr., Highland Park, Mich. (*Reinstatement*).	G. H. Cummins W. H. Baldwin	E. H. Clark M. B. Shea
WILLIAMS, A. E., Sales Engr., Buffalo Forge Co., Kansas City, Mo.	C. C. Cheyney W. R. Heath	Roswell Farnham A. A. Criqui